MACHINERY.

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SOME NEW FORMS OF TRANSMISSION DYNAMOMETER.

John J. Flather.

THE writer has been very much interested in the articles on Transmission Dynamometers which Mr. Webber has so ably presented in your columns, and it has occurred to him that possibly the readers of Machinery might be interested in a type of dynamometer entirely different from those described, which has proven very satisfactory under a variety of conditions in the laboratory and shop.

While investigating the subject of power transmission, as applied to milling machines, the writer constructed an apparatus several years ago, shown in Fig. 1, page 368, by which he proposed to measure the magnitude of the force exerted by the teeth of the cutter, but the results were not wholly satisfactory when applied to a milling machine. Used on a planer, however, a measure of the useful work was readily obtained from the card taken from the indicator attached.

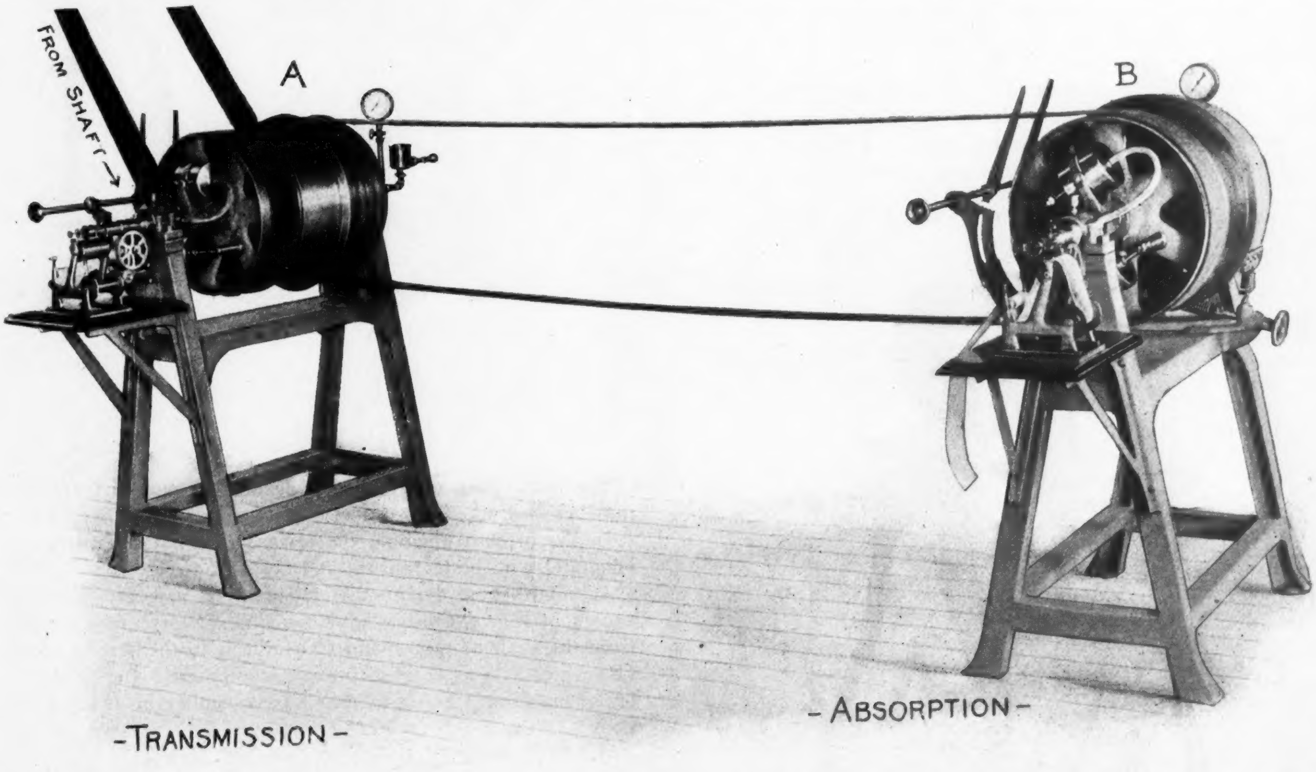


FIG. 4. TWO FLATHER DYNAMOMETERS, IDENTICAL IN EVERY RESPECT. ONE IS TRANSMITTING THE POWER FROM THE SHAFT, THE OTHER ABSORBING IT.

Prof. L. P. Breckenridge, of the University of Illinois, had previously made some interesting experiments with a similar apparatus for determining the pressure exerted by a drill working under known conditions, and later had very successfully applied the apparatus to planer tools. (See "American Machinist," Aug. 14, 1890.) The action of the apparatus will be understood by an inspection of the figure. A cylinder of known area is filled with oil and is provided with a pressure gauge and steam engine indicator, as shown. A plunger is fitted into the cylinder and the thrust of the tool forces it into the cylinder.

By a suitable arrangement of cords, the drum of the indicator is made to revolve synchronously with the stroke of the tool or with the work, and as the pencil is forced upward by the pressure exerted at the point of the tool, it will be seen that a measure of the work performed can be obtained from the card. The gauge is simply a check on the indicator. It is evident that the total work performed cannot be obtained by this means, as the force required to drive the machine itself is disregarded.

To obtain the total work, and at the same time the useful effect, the plan was adopted of mounting the cylinder upon a rotating pulley and forcing the oil through the center of the shaft, but in order to balance the pulleys two cylinders were used as shown in Fig. 2.

The pulley L which receives the driving belt is loose on the shaft and free to turn within certain limits. F, secured to the shaft, is belted to the machine to be tested, and carries a pair of cylinders C which are supported on trunnions; these cylinders are partially filled with oil, and connected to center of shaft by a small flexible tube. The end of the shaft is bored out and provided with a hollow steel tube free to revolve in the spindle and fitted with gland and stuffing-box nut to prevent leakage of oil. Connected with this steel tube is the stand S carrying a pressure-gauge and an indicator.

When motion is given to the pulley L it revolves through a small arc until a steel ball-pin in its arm comes in contact with the plunger in cylinder C. If there is no resistance to be overcome, the indicator-pencil and gauge-finger remain at zero; but as soon as resistance occurs the plunger is forced inwards. When the power overcomes the resistance motion is communicated to the pulley F, and the machine is driven through the force transmitted by the oil.

As the plunger is forced inwards, the indicator-pencil and gauge-finger will in consequence rise, and the amount of rise will determine the pressure per square inch acting on the plunger.

Figs. 1 and 6 in this article are from Prof. Flather's "Dynamometers and the Transmission of Power," and are used with the permission of the publishers, John Wiley & Sons, New York.—Editor.

If the distance of the lever-arm of the plunger is known, the power can be readily ascertained from the formula—

$$\text{H.-P.} = \frac{PV}{33000} = \frac{P \times 2\pi rN}{12 \times 33000},$$

in which r equals radius in inches of path traversed, and N equals number of revolutions per minute.

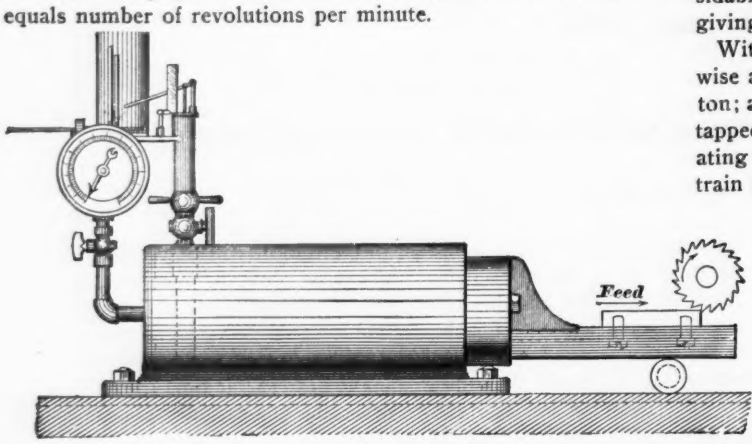


FIG. 1. THE ORIGINAL MACHINE.

In the experimental machine first constructed, the pulleys were each 12 inches in diameter and 3½ inches face; the cylinders were 1.954 inches in diameter, presenting an area of 3 square inches. The plungers were of hard bronze, and were kept tight by leather cup-washers secured to the end; grooves in the plunger, as for water-packing, and cast-iron piston rings were successively used, but abandoned after a short trial as not being trustworthy. A 5-pound spring was used in the indicator (a Tabor), as with stronger springs the card obtained was not sufficiently large to show the small differences of power which it was desired to bring out with certain machines.

An examination of an indicator card, Fig. 6, page 369, from this dynamometer shows that the power required to drive a 16-

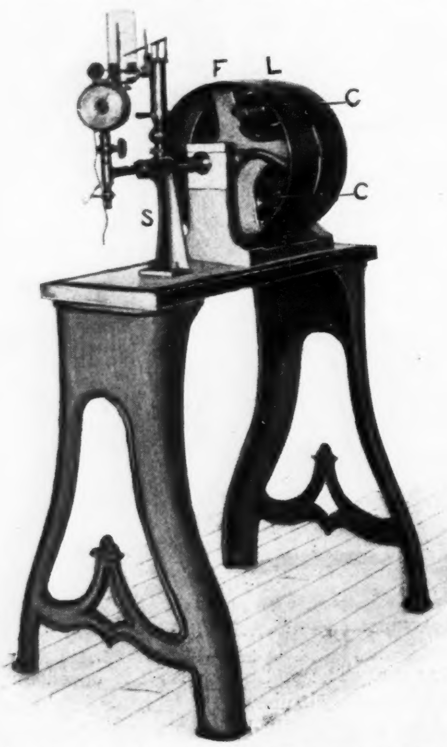


FIG. 2. EARLY FORM OF DYMAMOMETER.

inch Flather lathe with back gears in, well lubricated and running light at 36 revolutions per minute, is

$$\text{H.-P.} = \frac{PV}{33000} = \frac{0.11 \times 2\ (5 \times 3 \text{ sq. in.}) \times 2\pi \times 3.6'' \times 140}{33000 \times 12} =$$

.026 HP. = 868 foot-pounds.

where 0.11 equals the height of ordinate in inches; 5 pounds equals the spring used; area of each cylinder-piston equals 3 inches; radius of arms equals 3.6 inches; the revolutions of dynamometer being 140 per minute. With the screw feed in, still running light, the power was found to be 0.04 horse-power; with the load on, which was a light cut $\frac{1}{16}$ inch deep on a round bar of wrought-iron with diamond-pointed tool, the maximum power registered was 0.20 horse-power.

The machine just described enables one to use any make of indicator by attaching it to the stand, S, but it was thought desidable to make the dynamometer self-contained and capable of giving a continuous record.

With this object in view, the shaft was drilled through lengthwise and bored out at one end to fit a one-half square-inch piston; at the other end the bearing was fitted with a cap which was tapped to receive a pressure gauge and oil pump. At the operating end a worm was cut on the shaft which set in motion a train of gearing connecting the take-off mechanism.

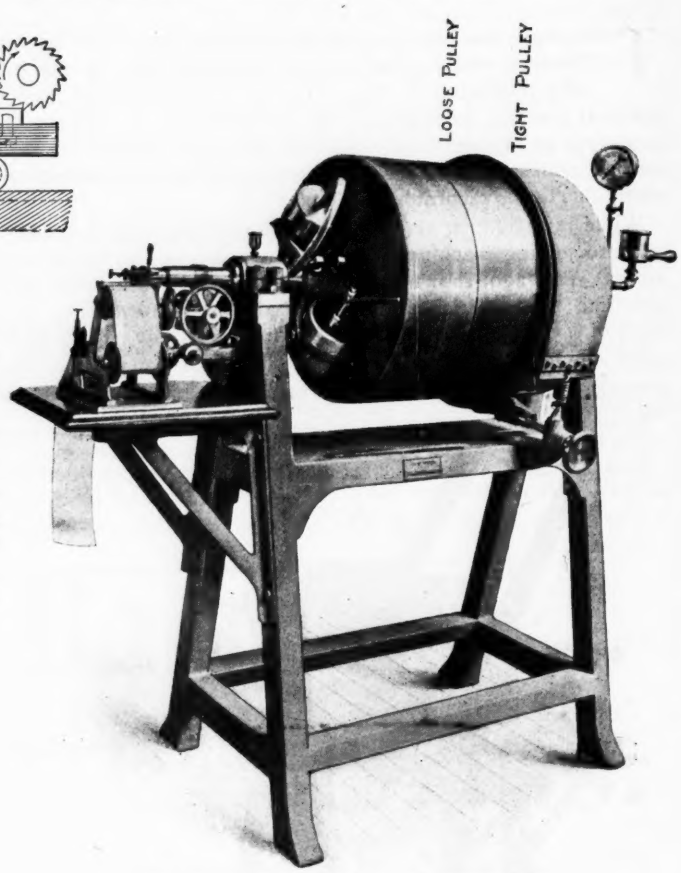


FIG. 3. LATEST FORM OF DYNAMOMETER.

The latest form of Flather Hydraulic Dynamometer embodying these features is shown in Figs. 3, page 368, and 4, page 367.

In the present machine the feature of greatest interest is its adaptability to both transmit and absorb power.

A pair of these dynamometers, identical in every respect, are shown in Fig. 4, the left hand machine (A) receives the power from the shaft or motor and transmits it to the second machine (B) which acts as an absorption dynamometer—the driving pulley of machine A being used as a brake pulley on machine B, as shown more clearly in Fig. 3.

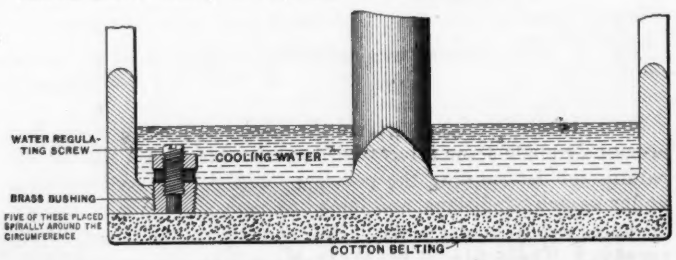


FIG. 5. DETAIL OF RIM OF FRICTION WHEEL.

This driving pulley is provided with internal flanges for carrying the cooling water, and the rim is furnished with a number of adjustable outlets as shown in Fig. 5, by which the quantity of water admitted to the brake band is readily controlled. The cooling water is supplied directly to the wheel from a small 4x9-inch circular tank attached to the machine, and provided with a regulating valve.

The brake-band is a 3-ply cotton belt suitably connected,

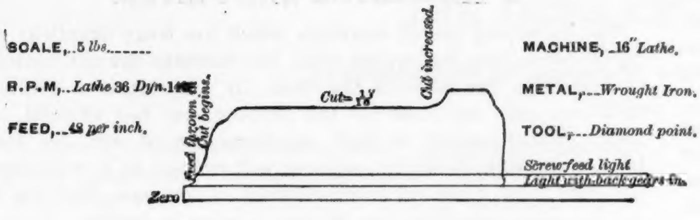


FIG. 6. DYNAMOMETER INDICATOR CARD.

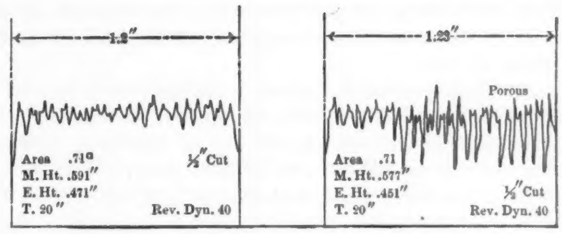


FIG. 7. TURNING CAST IRON—EFFECT OF BLOW HOLES.

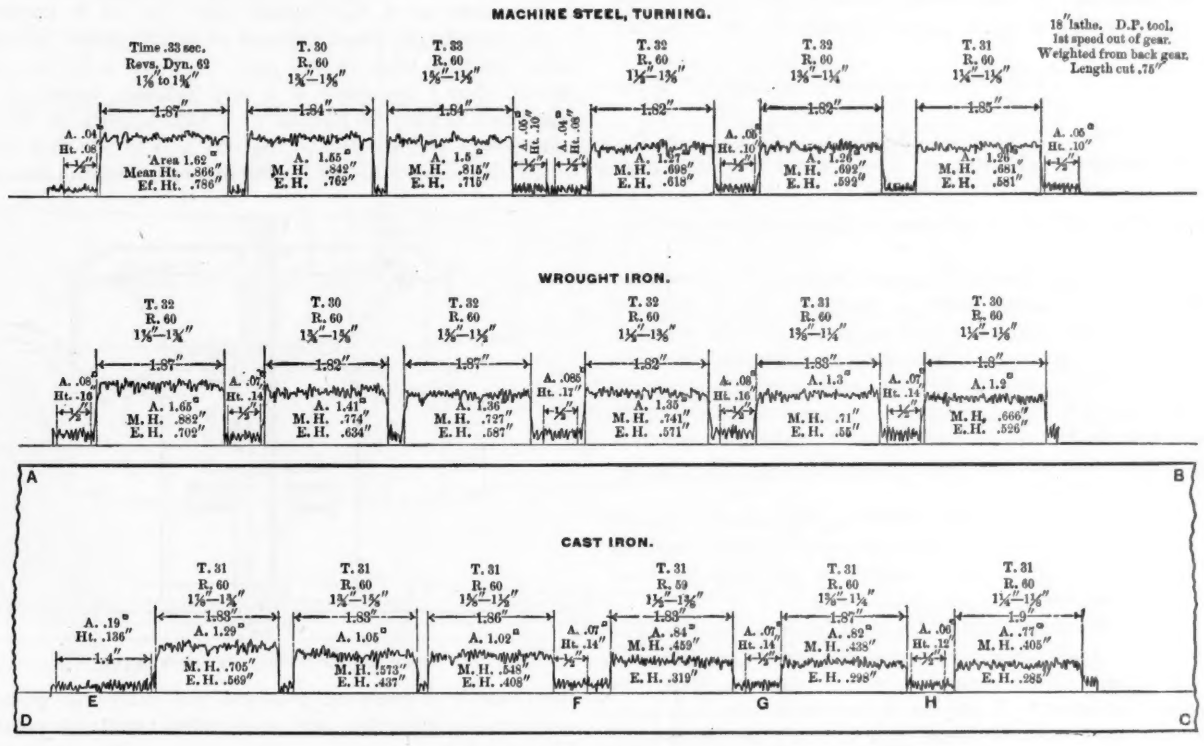


FIG. 8. TURNING DIFFERENT METALS.

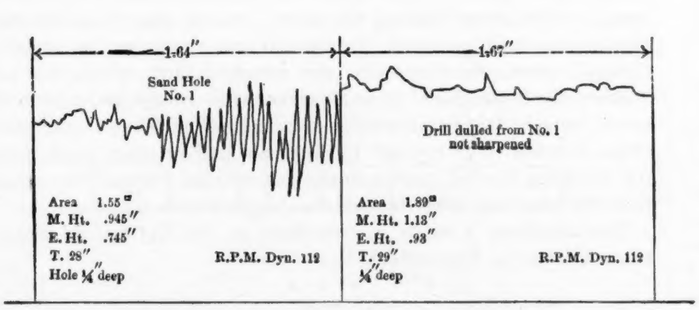


FIG. 9. DRILLING CAST IRON, 1-INCH HOLE.

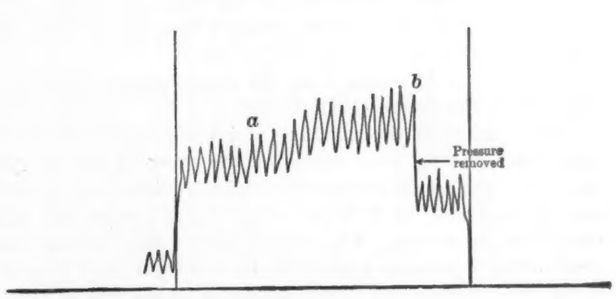


FIG. 10. EFFECT OF TIGHT CENTER ON LATHE.

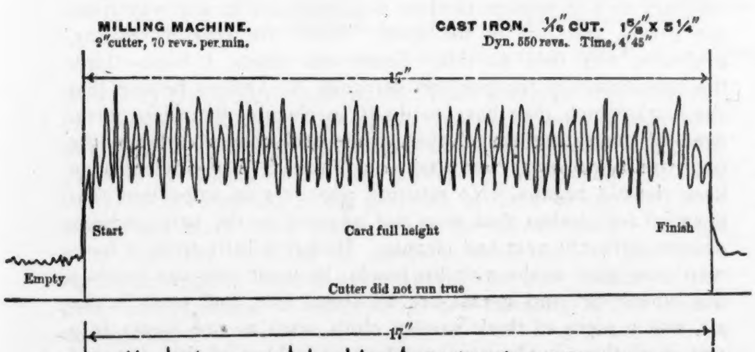


FIG. 11. CARDS FROM MILLING MACHINE.

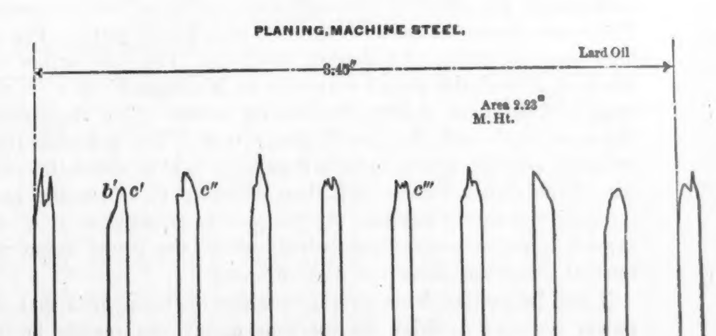


FIG. 12. CARDS FROM SHAPER—PLANING MACHINE STEEL.

through two sets of tension springs, to the brake-regulating screw. It will be seen that the brake can be readily removed, thus converting the machine into a transmission dynamometer.

In either case an automatic record is obtained from the indicator motion.

As the importance of power measurement in its various phases has become more generally recognized, the demand for a larger machine has also increased. In its improved form this dynamometer is furnished with 18-inch pulleys for a 6-inch double belt; it has been found desirable to turn the pulleys both inside and out, so they will be thoroughly balanced at high speeds. Each of the two cylinders has an area of 10 square inches, so that the pressure per square inch is never very great. The radius of the point of application of the force is 3.82 inches; in one revolution, then, the distance passed through by the actuating force is $\frac{2\pi r}{12} = 2$ feet. The horse-power transmitted or absorbed is, therefore, — H.-P. $= \frac{2PN}{33000}$, where P = total pressure in pounds, and N = revolutions of dynamometer per minute.

As in the other machines of this type the pressure is transmitted to the indicator piston which is furnished with a standard indicator spring, the scale of which depends upon the work to be done; thus for light work a light spring (5 pounds) would be used; on the other hand, for heavier work a 20-pound spring would be employed. The springs are changed as in any ordinary indicator; in fact, the indicators used in these dynamometers are regular standard makes, modified to suit the requirements of the machine.

In addition to the force curve traced by the indicator pencil, a zero or datum line is traced by another pencil, which may be located at pleasure. Thus one is enabled to ascentain the total work or any subdivided portion of it.

Mr, J. D. Hoffman, of Purdue University, constructed one of these hydraulic dynamometers about three years ago, concerning which he presented a paper before the American Society of Mechanical Engineers.* It may be of interest to note here some of the results which Mr. Hoffman gave at that time, illustrating the action of the dynamometer.

"Fig. 8 is taken from a tracing and serves to show the gradual reduction in height of the card as the diameter of the stock decreases; also, to compare the relative heights of cards for the different metals.

"The card as it comes from the dynamometer is four inches wide, shown at A, B, C, D.

"The friction of the machine and work was such a variable quantity, especially so considering the pressure on the tail-center, that it was found necessary to take friction cards at intervals during the series, as F G H. Figs. 8 and 9 show the effect of sand holes in casting. Fig. 10 was taken while turning machine steel. The tail-center proved to be too tight, and from a to b shows an increase, due to the grooving of the center. At b the pressure was removed.

"Fig. 11 shows a cut taken on the milling machine. The upper card shows the effect of running with the cutter out of center. The lower shows the same work done with a true cutter. Fig. 12 illustrates the action of a shaping machine. The tool strikes the work at a′ and the pencil responds to b′; from b′ to c′ is the length of the card during the cutting action. The tool leaves the work at c′, and the pencil drops to d′. The machine then reverses, and the pencil mounts gradually to e′ at about the middle of the return stroke, and then down to f′, where the next forward movement begins. As the tool is released at c′ c″ c‴ there is apparently a reaction which carries the pencil below the normal condition, shown at d′ d″ d‴, etc."

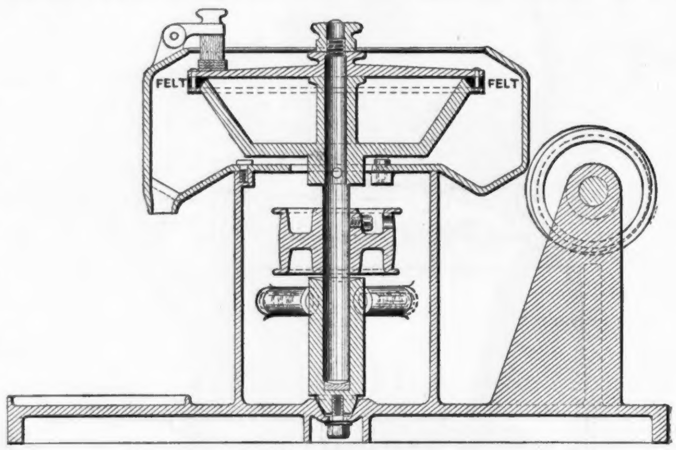
It will be noticed from an examination of these cards that the power required to drive the machine empty can readily be determined if desired, for, all that is necessary is to run the machine at the same speed without any pressure on the cutting tool, and the resulting card will give a measure of the work. The same is true of the power required to operate the feed mechanism of the machine, or the dynamometer itself.

* American Society Mechanical Engineers Trans., Vol. XVII.

A SEPARATOR AND FILTER.

The increasing use of machines which use large quantities of oil for lubricating the cutting tools, has rendered the old method of separating the oil from the chips, by draining in a settling tank, not only too slow for the purpose, but too wasteful, as well. Where several of such machines are in use, the loss from separating oil by this method will amount to a considerable item; and if the oil is not filtered at frequent intervals, it will become so impure as to be very inferior in quality.

The centrifugal separator shown herewith is designed to both separate and filter the oil from 300 pounds of chips per hour. It consists essentially of a pan of the shape shown, holding from 30 to 45 pounds of chips, according to the size of the machine. It is rotated at a high speed, and the oil is thrown from it by centrifugal force through a small space between the cover and the body of the pan. The pan is at the top of a vertical shaft, supported in a step bearing, which in turn is supported flexibly by means of a ball bearing at the bottom and springs at the sides, so that the axis of the shaft is free to follow the motion of the center of gravity of the pan and the chips which it contains, provided this center of gravity does not coincide with this axis, as is quite certain to be the case. This condition insures smooth running. The arrangement of the belt pulleys is clearly shown.



SECTIONAL VIEW OF SEPARATOR.

The filtering of the oil is accomplished by a strip of felt fastened to the outer edge of the cover, and as the oil leaves the pan at the top it is forced through the felt by the action of centrifugal force, the impurities are retained, and when the oil reaches the outer bowl it is as clean as it would have been if extra time had been taken for filtering. The base of the machine is made large enough to receive a pail or other receptacle for catching the oil, and a brake is provided for quickly stopping the machine after the belt has been thrown off.

This machine is made in two sizes by the Springfield Separator Company, Springfield, Vt.

* * *

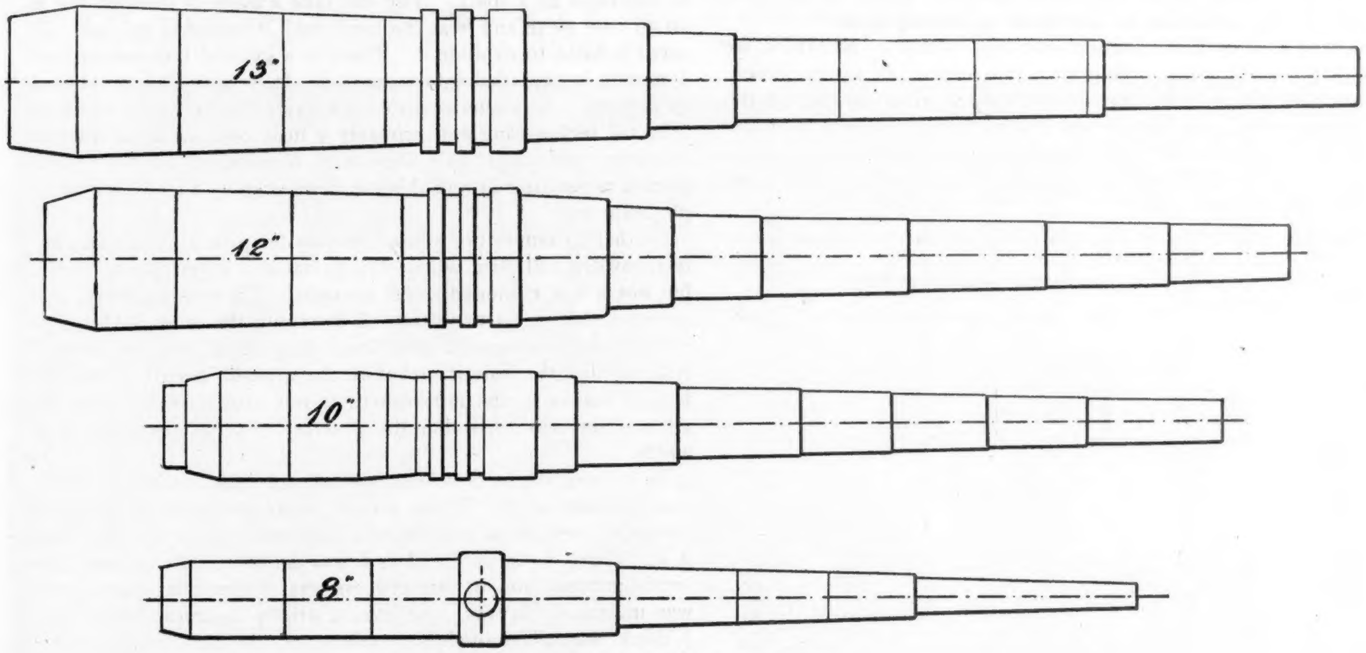
Some mechanics, when they are promoted to the position of foreman, discard the overalls and jumper and go about in an ordinary suit of woolen clothes, not protected in any way from the grease and dirt of the shop. There are several reasons, probably, why they do this. Some—not many, I hope—think the importance of the position demands it. Others believe that the work which they have to do is so cleanly that there is no need of protecting the clothes, while still others embrace the opportunity to wear out their old clothes. My advice is to keep the old regalia. No position was ever so important that it called for clothes that were not adapted to the surroundings or that were not neat and cleanly. However little work a foreman may have to do with his hands, he must stay ten hours a day where dirt and grease are all about him, and even in the air, and a piece of thick woolen cloth, such as one wears in a suit of clothes, is about as good an absorbent of this dirt and grease as can be made. How do you think a white woolen suit would look after it had been worn about the shop for a few months? It would get no dirtier than the others in the same length of time. Better give the old clothes to some one who needs them, or at least cover them up with the old overalls and jumper. OLIN SNOW.

THE GUNS OF OUR NAVY.

INFORMATION OF INTEREST, TECHNICAL AND OTHERWISE.

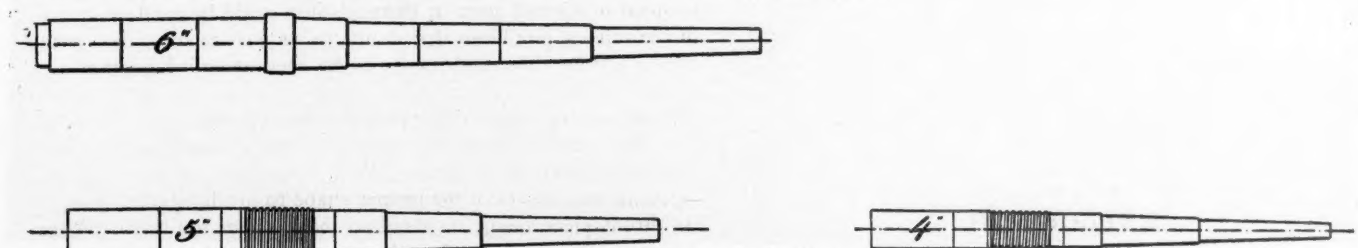
HENRY HESS.

In these days much attention is drawn to ships of war and their fighting equipment. The latter particularly excites the interest of the average landsman, especially that of the mechanic, and deservedly so, for guns, large and small, are magnificent specimens of the machinist's art and well calculated to delight the eye of those who can appreciate the ingenious mechanical contrivances and novel movements of the various breech mechanisms and the magnificent workmanship of the entire gun.



GUNS OF THE MAIN BATTERY.

As employed on board ship, guns are divided into two principal classes—the main and secondary batteries. To the main armament belong the heavy guns from 8 inches up. The secondary armament takes in all the lighter guns, and is again subdivided into rapid fire and machine guns. The rapid fire guns range from one pounder (about 1½ inch) up to 6-inch bore; the machine guns include the smaller guns, such as the Gatling, Revolver Hotchkiss, Gardner and Maxim-Nordenfeldt. The rapid fire guns, especially as regards the larger calibres, are of com-



4-INCH, 5-INCH AND 6 INCH GUNS OF THE SECONDARY BATTERY.

paratively recent introduction; their chief characteristic is the use of "fixed ammunition;" this means that the powder is contained in a metallic shell, which frequently carries the projectile as well; the whole is simply a very large brother of the familiar rifle cartridge. This extension of the principle of fixed ammunition permits much more rapid firing, hence the name, because it does away with the necessity of carefully swabbing the gun after each discharge to remove every spark, and also because of the great simplification of construction and handling of the breech mechanism.

The length of a gun is generally stated in terms of the calibre; most of our naval guns vary between 35 and 40 calibre; but Canet, the most noted French builder of ordnance, has constructed guns up to 60 calibre. The advantage of a long gun is due to the fact that the powder-gas acts in the projectile for a longer time, and so for a given greatest pressure stirs up more energy in the shot, or, on the other hand, permits the use of a lower maximum pressure without loss of energy. But excessively long guns have a tendency to droop at the muzzle, while, in naval work especially, this great length makes them far more vulnerable to the enemy's fire, as well as awkward to handle and house.

The sketches show to the same scale the various guns of the main and secondary battery of our 4, 5, 6, 8, 10, 12 and 13-inch calibre. One characteristic of these various guns that at once catches the eye is the arrangement of the trunnion seat. The 4 and 5-inch have a long thread for the reception of a nut that is attached to the recoil controlling mechanism and slides in guides that carry the trunnions. The 6 and 8-inch rifles have the trunnions directly attached, while the larger guns are provided with grooves for the reception of heavy straps, attaching them to the mount; trunnions and recoil arrangements are provided in the carriage.

The principal elements of typical guns of each size are given in the table below.

Calibres, inches.	Length, feet.	Weight in lbs. Gun.	Weight in lbs. Powder.	Weight in lbs. Projectile	Ratio of gun and shot, weight.	Greatest pressure, lbs. to inch.	Muzzle velocity, foot seconds.	Muzzle energy, foot tons.	Penetration of steel at muzzle, inches.
4	13.7	3,400	14	33	1:103	30,000	2,000	915	7.18
5	17.4	7,000	30	50	1:140	"	2,050	1,754	9.00
6	18.8	11,554	47	100	1:116	"	2,080	2,290	10.86
8	25.4	27,400	115	250	1:118	"	2,080	7,498	15.61
10	31.2	61,900	240	500	1:124	"	2,100	15,285	24.16
12	36.8	101.300	425	850	1:119	"	2,100	25,985	26.10
13	40.0	135,500	550	1,100	1:123	"	2,100	33,627	26.68

A glance at a list of figures in a column does not impress one with their real significance. Take f. i., the muzzle velocity; the table says it is 2,100 feet per second for a 12-inch gun; every one knows that means the shot is striking a fairly lively gait out of that muzzle in its anxiety to get away from the powder burning its rear; but when, after a little figuring, it is found that 2,100 feet per second means 1,430 miles an hour, or about 24 times as

fast as a mile-a-minute express train, he rather gets the notion that the "cannon ball train" is not in the running with its godfather.

The 12-inch gun has a muzzle velocity of 25.985 foot-tons per second; a train running at 60 miles per hour would have to weigh 360 tons in order to have the same energy; 200 tons is a pretty fair load for a locomotive of that speed.

And yet the builders of guns are continually striving to get more out of their guns. An English concern, Vicker's Sons & Maxim, make a 12-inch gun weighing 113,000 pounds that imparts to an 850-pound shot the enormous muzzle velocity of 2,750 feet per second; it will penetrate 35.5 inches of steel at the muzzle; this gun uses a smokeless cordite powder that develops a pressure of 38,000 pounds per square inch. Our navy still uses the ordinary black and cocoa powders, which do not give quite as high velocities as the newer smokeless ones.

[As of interest in connection with this article by Mr. Hess, we append the following table giving the number of 4-inch, 5-inch, 6-inch, 8-inch, 10-inch, 12-inch and 13-inch guns carried by the leading vessels in the United States Navy. Different tables that we have been able to obtain vary somewhat, and we have relied largely on the one published by the *Scientific American.* —EDITOR.]

TABLE OF ARMAMENT OF LEADING SHIPS OF UNITED STATES NAVY.

NAME OF VESSEL.	NUMBER AND SIZE OF GUNS.						
	4-inch.	5-inch.	6-inch.	8-inch.	10-inch.	12-inch.	13-inch.
Amphitrite	2				4		
Atlanta			6	2			
Baltimore			6	4			
Bennington			6				
Boston			6	2			
Brooklyn		12		8			
Castine	8						
Charleston			6	2			
Chicago		2	8	4			
Cincinnati		10	1				
Columbia	8		2	1			
Concord			6				
Detroit		9					
Dixie		10					
Harvard		6					
Helena	8						
Indiana			4	8			4
Iowa	6			8		4	
Machias	8						
Marblehead		9					
Massachusetts			4	8			4
Miantonomah					4		
Minneapolis	8		2	1			
Monadnock	2				4		
Monterey					2	2	
Montgomery		9	2				
Nashville	8						
Newark			12				
New Orleans		Four 5.7 inch.	6				
New York	12			6			
Olympia		10		4			
Oregon			4	8			4
Philadelphia			12				
Prairie		10					
Puritan	6					4	
Raleigh		10	1				
San Francisco			12				
St. Louis		4					
St. Paul		6					
Terror					4		
Texas			6			2	
Wilmington	8						
Yale		6					
Yankee		10					
Yorktown			6				
Yosemite		10					

* * *

EXPLOSIVES.

At a recent meeting of the Engineers' Society of Western Pennsylvania, the subject of explosives was up for discussion and several interesting points were brought out. Mr. Arthur Kirk opened the meeting by reading a paper giving the history of the advance and use of explosives. The following disconnected notes are taken from this paper and from the discussion which followed by different members, and include some of the more interesting facts that were mentioned.

Explosives, as now used, are of far more importance to the comforts of civilized life than most persons have any knowledge of. Without explosives, we could not make iron or steel; or a single piece of glass; or build a substantial building of any kind; or a railroad or canal. Let the reader stop a moment and try to imagine a state of society without explosives to get out coal, iron ore, lime-stone, building stone, and fire clay. Gold and silver mines and railroads would be unknown. This, of course, would carry with it the total annihilation of all iron and steel tools and machines; our saws, hatchets and axes would at once disappear, and we would step away back to the stone age. There could be no substantial building erected without powder to blast stone for the foundation, and the greater part of all our red or fire brick is now made from blasted material.

Formerly dynamite was much more easily exploded than it is now. Dynamite is now mixed with nitrate of soda, and in the stick form, as it is now prepared, many people consider it as harmless to handle as so much wood. I remember going out to use it at a quarry about 50 feet deep down, straight as a wall, to where the men were working. I told them I was going to throw down the dynamite, and it did not take them a second to get out of the way. In throwing the dynamite down, some of the cartridges were broken, but I picked them up and did excellent execution,

Black powder is exploded by a flame, but dynamite can only be exploded by a spark. You can take a piece of powder, lay it on an iron sieve and heat the sieve until it becomes red hot. It needs a flame to explode it. There is a general impression that dynamite is exploded by concussion. It is not. It is exploded by a spark. It is almost invariably exploded by a cap which is fully 1¼ inches long and probably a little over ¼ of an inch in diameter, and about half filled with fulminating mercury, and electric wires are run into this cap connected by a small bridge of platinum wire.

Neither dynamite nor smokeless powder give a flame. Smokeless powder will burn with a bright flame if given plenty of air, but not if it is exploded under pressure. There is a general impression that in an explosion of dynamite the greatest force is exerted in a downward direction. Regarding this question, I will say that the hole it makes in the ground is visible, but the hole it makes in the atmosphere is not visible, and hence the general impression is that the greatest force is exerted downwards.

As to smokeless powders, several opinions were expressed. One speaker said: "There are so many secret processes and formulas used in its manufacture, that one can hardly tell what it is. When I was in Sweden, I was urged to take charge of a secret process, and in this process one of the chief ingredients was molasses. It was, I believe, a strictly chemical compound. I think that all smokeless powders are chemical compounds." Another speaker said: "I think the majority of smokeless powders are composed of mixtures of gun cotton and nitro-glycerine.

* * *

MAKING WOODEN POLES.

Round wooden poles, 9 to 12 feet long and 2 to 2¾ inches in diameter, can be easily and cheaply produced in a shop properly fitted up for them; but to make them in lots of a few dozen only, to sell from 69 cents to $1 each, all finished in shellac and polished, in a shop equipped with only the ordinary pattern makers' machinery, is quite another problem, and the method adopted in a small shop in Pennsylvania could be used on many of the jobs of moulding that come up, as well as for poles, even if it is a little risky and hardly to be recommended on that account.

First, square strips of the proper size were sawn out, then the saw was removed from the saw arbor and a square cutter head similar to that in a "buzz planer" was put on in its place, carrying two knives of the proper shape to produce a half-round, its throat piece being removed to make room for them. After adjusting the guide, each strip was run over the cutters twice, making it almost perfectly round and straight. Then the strips were put into a lathe and any little rib that the knives may have left was removed with a plane or spoke shave, after which they were sandpapered and polished. In running the strips over the knives the second time, of course they were turned end for end, instead of being rolled over, so that the same side of the strip lay against the guide, making any little variation in its width less liable to produce a pole without the opposite rounds overlapping. By the method described above fifty 12-foot maple poles, 2¾ inches in diameter were made, and it is claimed that the time required averaged thirty minutes per pole, which was about one-third of the time needed to turn them in the same shop, not including the shellacing or polishing in either case, nor the time used in making the special cutter head. A board with sufficient spring to hold the strips down on to the cutters was clamped to the saw table, and although the poles were not absolutely round they were good enough for all ordinary purposes when done. MILO.

A NOVEL DINNER PARTY

AND SOME NOTES FROM THE PLACE WHERE IT OCCURRED.

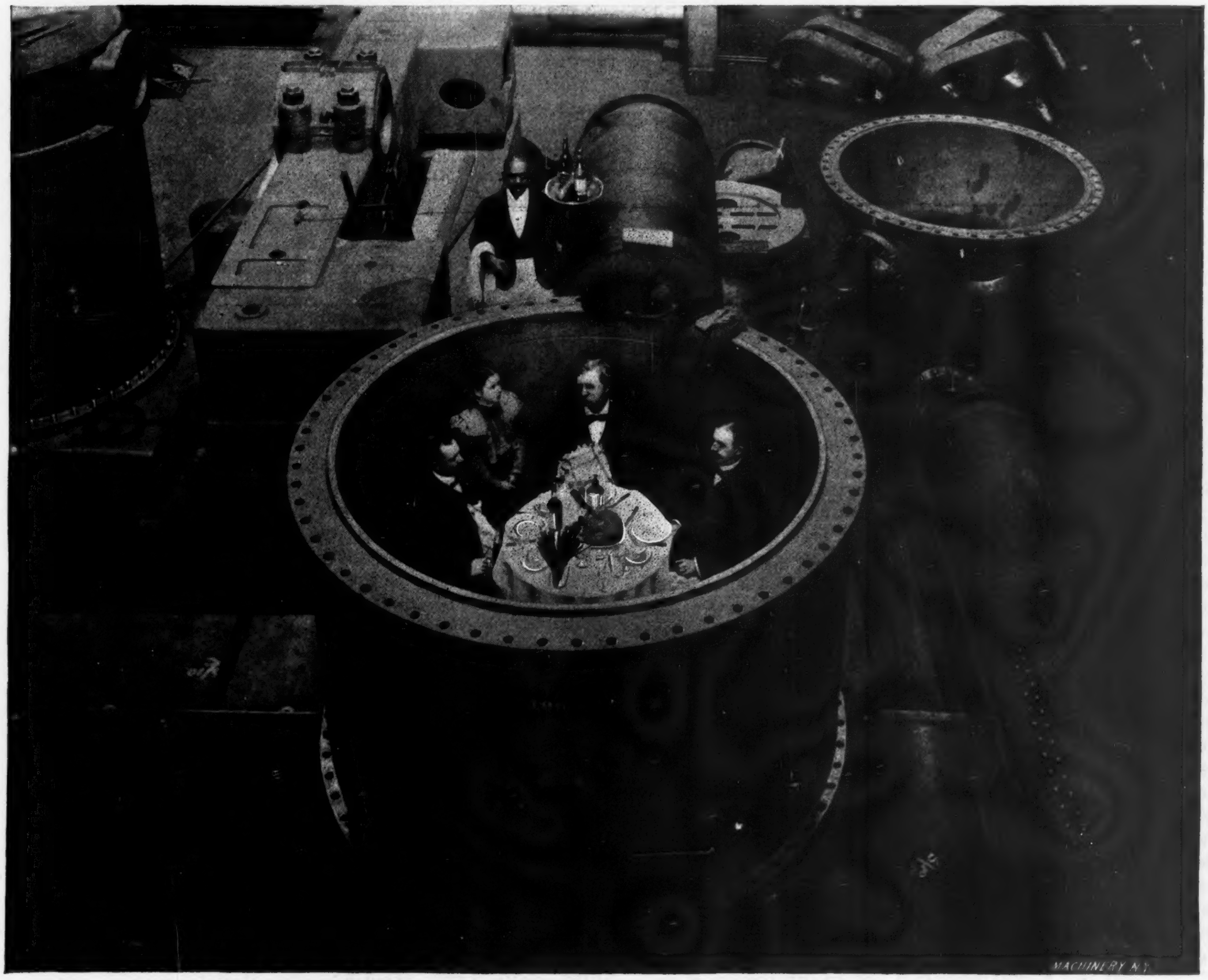
The accompanying illustration is from a photograph and shows one of the most novel, and doubtless one of the most enjoyable, dinner parties that it was ever the good fortune of the participants in this one to indulge in. The repast was served by the Lake Erie Engineering Works, Buffalo, N. Y. The table was set in the low pressure steam cylinder of a triple-expansion pumping engine built by these works for the Chicago Water Works, before the engine had been shipped from the shop. Superintendent P. H. Kane, of the Lake Erie Company, acted as host, and the guests were G. B. Bartlett, of Robt. W. Hunt & Co., Chicago; Miss Florence Bartlett and D. W. Lanagan, chief engineer of the pumping stations of Chicago, Ill.

While sixty inches has the call
As up, and down, we rank it.

Old Hercules made a river wind
To help his Augean labor—
Chicago beats Old "Herky" blind
She pumps a lake—her neighbor.

It is remarked by Mr. Lanagan that when he reached the word "banquet" in the second verse, he thought that he was floored, and that it would be a hopeless case trying to make a word rhyme with it; but he thinks the way he got out of the trouble at the end of the last line of that verse entitles him to a place among the poets.

The engine for which this cylinder was made is 30,000,000 gallons capacity in twenty-four hours, and is now being erected at the Fourteenth street pumping station at Chicago, where there are already three Allis engines of 15,000,000 gallons' ca-



A LARGE CYLINDER AND A GOOD DINNER.—CYLINDER 94 INCHES IN DIAMETER.

The occasion was a visit to the Lake Erie Works last May by these gentlemen and Miss Bartlett, Mr. Bartlett coming in the capacity of inspector. The unusual size of the low-pressure steam cylinder suggested to Mr. Lanagan the happy idea of the dinner party, which was at once acted upon with a generous spirit of hospitality. Mr. Lanagan seems to have been the inspired one of the party and to have gone to Buffalo in a very prolific frame of mind, for he developed a poetical vein and composed the following verses upon the spur of the moment, which tell several facts about the engine.

Circles are trumps, in this brave show
Of Cylinder girth, and table,
Lake Erie's Works make these things go
Because they're fit and able.

It's seven feet ten, from wall to wall
Surrounding group, and banquet,

pacity each. It is to pump directly into the mains from the four-mile crib extending into the lake. The engine is six feet stroke and the high, intermediate and low-pressure cylinders are respectively 34", 63" and 94" in diameter. The liner and casing of the low-pressure cylinder weighs 38,000 pounds and the cylinder heads 28,000 pounds each. These heads are of peculiar construction, being designed to contain the valves, which are of the Corliss type, and it will readily be appreciated that to use a Corliss valve on a 94-inch cylinder requires a well thought out design and good construction to prevent both leakage and binding. The quality of work that has been put on these valves and their seats, however, bears conclusive evidence that these points have not been lost sight of. The flywheels of this engine are 20 feet in diameter and weigh 44 tons each.

Concerning the Lake Erie Works.

The Lake Erie Engineering Works have a shop well equipped with large tools designed for the heaviest class of work. They

are fortunate in possessing a shop with head room enough under the traveling crane to erect large vertical engines complete in the shop and thus save the annoyances and expense that always arise when work cannot be put together before shipment. Among the large tools, perhaps the most interesting is a 28-foot extension boring mill on which the low-pressure cylinder above mentioned was bored and faced. This mill is of special construction, specially designed by themselves, embodying several ideas of Superintendent Kane, who is a believer in power and rigidity in machine tools to the fullest extent. The table of this mill is 12 feet in diameter, so large, in fact, that when it came to the shipment of it, a hole had to be cut in the bottom of a platform car, and the table, standing on edge, was let through this opening, to within 2 feet of the track. The table was too large to ship flatwise. Not the least important or interesting part of the table, however, is the spindle, which is 18 feet long, 36 inches diameter at the top and 18 inches at the bottom.

A back tool post is placed on the back part of the main bed plate of the mill, consisting of a stiff, upright column having a slide on which the tool block can be moved up or down. This makes a very stiff arrangement for turning large pieces on the outside, and at the same time leaves the heads free for inside work or facing.

One of the first jobs that was done on this boring mill was the barbettes for the United States cruiser New York, which is now Rear Admiral Wm. T. Sampson's flag ship. They were 24 feet in diameter, 14 inches thick, and 8 feet 6 inches high, and of Harveyized steel. These had to be turned outside and inside and, to begin with, were not very easy pieces to hold on a 12-foot table. The greatest trouble, however, was encountered in getting steel for the tools that would stand, and in learning what kind of cuts to take. After much experimenting, Mr. Kane found that he secured the best results with an English self-hardening brand, and by taking thin, broad cuts, almost like strips of paper.

While these barbettes were a notable piece of work, the large cylinder for the pumping engine is not by any means to be despised as a machine job. In general we believe that mechanics would hold that a cylinder ought to be finished with a single tool to insure a smooth and nicely lapped cut, but in this case nine tools were used for both roughing and finishing. A special 12-inch boring bar was fitted up with a cutter head carrying the nine tools and the result was all that could be desired, and when the cylinder came off the mill it did not vary .002" in diameter from top to bottom, nor in any of the directions of its diameter. It was a good piece of work.

* * *.

A prominent machinery man, who has watched the course of events for many years, says that it is useless to try to sell high-grade machinery where the users have been accustomed to a very low grade. He says, for example, that it does not pay to try to sell engines producing a superior economy where old traps have been in use for years, burning up coal by the carload. To such users an engine is an engine, and they do not appreciate the value of high-class "notions" for saving coal. Again, once get a good engine in a town, using, say, three pounds of coal, and the next man in that town who buys an engine will want to go him one better and have an engine using only 2½ pounds. A little education and rivalry thrown in are all that are needed to change one's views.

* * *

Once in a while a job comes along where a casting like a cylinder or a piston, must be made air and water tight. A test under water pressure will disclose the leaks, and if there is time enough the most effective remedy is to let the casting stand and rust up. It should be put under hydraulic pressure occasionally, both for the purpose of finding out how the rusting is progressing and to fill the pores with water and so hasten the rusting process. This action can be made more rapid by forcing in a solution of sal-ammoniac. Plugging and peneing are sometimes resorted to in stopping leaks, but they are of doubtful utility. If a casting is very porous, an effort to plug the hole is almost sure to disclose a leak somewhere else. One company, that has been making machines for years having cylinders that have to be air tight, always make them so by rusting, and seldom lose a cylinder.

THE DESIGN OF FAN BLOWERS*.

A DISCUSSION OF THE THEORY OF THESE BLOWERS, WITH THE NECESSARY CALCULATIONS FOR DETERMINING THEIR PROPORTIONS.

WALTER B SNOW.

According to the purpose for which they are designed, fan blowers may be classed either as volume blowers or pressure blowers, although one type naturally merges into the other. In either type the fan blower, proper, consists in its simplest form of a number of blades extending radially, or nearly so, from its axis and presenting practically flat surfaces to the air as they revolve. By the action of the wheel the air is drawn in axially at the center and delivered from the tips of the blades in a tangential direction. It is therefore designated as a centrifugal fan, or, more properly, as a peripheral discharge fan.

As will appear more clearly in what follows, a volume blower is primarily designed to discharge air in large quantity under low pressure with the minimum expenditure of power. This requires a wide and comparatively slow running wheel. A pressure blower, on the other hand, is designed for the purpose of creating a high pressure, which may be as great as 20 ounces per square inch, while delivering a relatively small volume of air. To this end the wheel must be narrow and operated at high speed.

In operation, the peripheral discharge fan sets in motion the air within it, which, acting by centrifugal force, is delivered tangentially at the outer circumference of the wheel. Air rushes in at the axial inlet to fill the space between the blades, in which there is, by the centrifugal action, a tendency to form a vacuum. The degree of this vacuum is dependent upon the circumferential speed of the wheel; and the velocity of the air discharged through an outlet of proper size is substantially equal to that of the circumference of the wheel. The fan case thus virtually becomes a reservoir from which the air escapes through the outlet, the proper size of which to produce the above stated results will be discussed further on.

The velocity with which air escapes into the atmosphere from a reservoir is dependent upon the pressure therein maintained and upon the density of the air. The pressure per unit of area divided by the density per unit of volume gives the head, usually designated as the "head due to the velocity." The velocity produced is that which would result if a body should fall freely through a distance equal to this head. In the case of the flow of water such a head always exists; as, for instance, when a standpipe is employed to produce the requisite pressure. Suppose the head of water to be 50 feet and its weight per cubic foot to be 62.5 pounds, then the pressure per square foot will be $50 \times 62.5 = 3{,}125$, and that per square inch $3{,}125 \div 144 = 21.7$ pounds. Its theoretical velocity of flow from an orifice at the bottom of the standpipe would be 56.7 feet per second, as determined by the formula for falling bodies, which is $V = \sqrt{2\,g\,h}$, in which

v = velocity in feet per second.
g = acceleration due to gravity.
h = head in feet, here 50 feet.

In the case of air, however, an actual homogeneous head never exists, but in its stead we have to deal with an ideal head which can only be determined by dividing the pressure by the density. As the density of air is so much less than that of water it is evident that for a given pressure the head will be far greater in the case of air. But the velocity of discharge is dependent only on the distance fallen which is represented by the head, whether real or ideal. As a consequence, air under a stated pressure escapes at vastly higher velocity than water under the same conditions.

A clearer comprehension of the meaning of an ideal head may be obtained by a consideration of the conditions existing under ordinary atmospheric pressure. The pressure of the atmosphere is due to the weight of a column of air, and for any given area is to be measured by the weight of a column of air having the given area as a base, and a height equal to that of the atmosphere. But this height cannot be accurately determined, and, furthermore, the density of the air decreases in geometric ratio as the distance from the earth increases. For the purposes of

* The body of this article is taken from a paper upon "The Uses of the Blower in the Foundry," read by Mr. Snow before the New England Foundrymen's Association. Mr. Snow has since added to this abstract, however, putting it in its present form for the benefit of the readers of MACHINERY.—[EDITOR.]

calculation, however, the practical equivalent of such a column may be determined by assuming the air to be of uniform density throughout, and the column of such a height as to weigh the same and to produce the same effective pressure per unit of area.

Under the standard conditions of barometric pressure of 29.921 inches the atmospheric pressure is 14.69 pounds per square inch, or 2,115.36 pounds per square foot. At this pressure a cubic foot of dry air at 50 degrees has a density of 0.077884 pounds. Consequently a homogeneous volume $\frac{2115.36}{0.077884} = 27{,}170$ feet high, having a base of 1 square foot area, would weigh 2115.36 pounds and exert this pressure upon the given area.

If air under this head were to be allowed to flow freely into a vacuum, the velocity, per the formula above quoted, would be

$$V = \sqrt{2g \times 27160}$$
$$= \sqrt{64.32 \times 27160}$$
$$= 1321.7 \text{ feet per second.}$$

In the case of air flowing into a vacuum, the head is actual, although here reduced, for simplicity, to that of a homogeneous head. But under any other conditions both the pressure head and the velocity head are purely ideal.

We may now return to a comparison of the velocity of water and air under the same pressure. For air under a pressure of 21.7 pounds per square inch = 3,125 pounds per square foot and at a temperature of 50 degrees, the equation becomes:

$$V = \sqrt{64.32 \times \frac{3125}{0.777884}}$$
$$= 1606.4 \text{ feet per second.}$$

That is to say, the velocity due to a given pressure, under these stated conditions is $\frac{1606.4}{56.7} = 28.3$ times as much in the case of air as in that of water.

In the attempt to force air at a given velocity through a given pipe, it is the province of the fan wheel, if employed therefor, to create within the fan case a total pressure above the atmosphere which shall be sufficient to produce the velocity and also overcome the resistance of the case and the pipe. If, however, the pipe be removed and the fan be allowed to discharge the air through a short and properly shaped outlet, the pressure necessary will, with an efficient fan, be substantially that required to produce the velocity. The method of determining the velocity due to any given pressure has just been explained. From the same formulae, properly transposed, the pressure due to any given velocity or necessary to its creation may be determined. The pressure thus determined is properly that which it is the purpose of a fan, employed as a device for moving air, to create.

The velocity of the fan tips or circumference of the fan wheel which is necessary to produce a given velocity of flow through a properly shaped outlet within the capacity of the fan, is substantially equal to the velocity of flow. If, therefore, the peripheral velocity of a given fan is known, the resulting pressure for the production of velocity through an outlet of proper size and shape may be readily calculated.

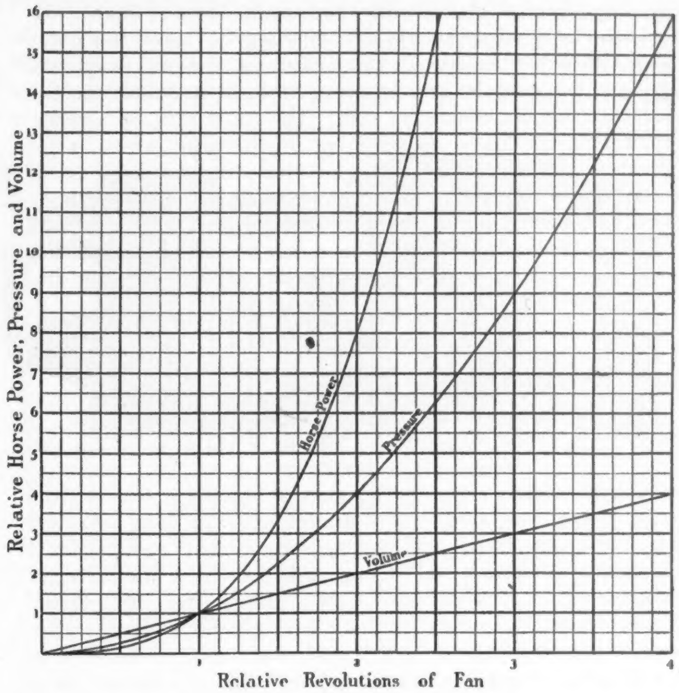
From the basis formula already employed, as well as from the preceding discussion, it is evident that the pressure created by a given fan varies as the square of its speed. That is, doubling the speed increases the pressure four-fold. The volume of air delivered is, however, practically constant per revolution, and therefore is directly proportional to the speed.

The work done by a fan in moving air is represented by the distance through which the total pressure is exerted in a given time. As ordinarily expressed in foot pounds, the work per second would, therefore, be the product of the velocity of the air in feet per second, the pressure in pounds per square foot, and the effective area in square feet over which the pressure is exerted.

From this it is evident that the work done varies as the cube of the velocity, or as the cube of the revolutions of the fan. That is, eight times the power is required at twice the speed. The reason is evident in the fact that the pressure increases as the square of the velocity, while the velocity itself coincidently increases; hence, the product of these two factors of the power required is indicated by the cube of the velocity.

The theoretical relations between the revolutions of a fan and the volume, pressure and horse-power are clearly shown by the accompanying diagram.

THEORETICAL RELATIONS BETWEEN THE REVOLUTIONS OF A FAN, THE VOLUME DISCHARGED, THE PRESSURE CREATED AND THE HORSE-POWER REQUIRED.



These curves are based upon the facts that the volume varies directly as the speed, the pressure as the square and the horsepower as the cube of the speed. Thus it is shown by the curves that if the speed is doubled the volume is also doubled, the pressure is increased four times and the horse-power becomes eight times greater. The tremendous power expenditure required for even a moderate increase of speed is thus rendered distinctly evident.

In selecting a fan, the facts just presented should be borne in mind. It appears to be so simple to secure increased volume by running a given fan at higher speed, that the influence upon the power required is frequently overlooked. If the necessary amount of power is actually furnished, its expenditure will entail great loss in efficiency as compared with that required to operate a fan properly proportioned to the work.

In the design of a wheel to meet given requirements it is necessary to make its peripheral speed such as to create the desired pressure, and then to so proportion its width as to provide for the required air volume. Evidently, the velocity and corresponding pressure may be obtained either with a small wheel running at high speed or a large wheel running at low speed. But, if the diameter of the wheel be taken too small, it may be impossible to adopt a width, within reasonable limits, which will permit of the passage of the necessary amount of air under the desired pressure. Under this condition it will be necessary to run the fan at higher speed in order to obtain the desired volume. But this results in raising the pressure above that desired, and in unnecessarily increasing the power required. On the other hand, if the wheel be made of excessive diameter, it will become more impracticable on account of its narrowness. Between these two extremes a diameter must be intelligently adopted that will give the best proportions for the specific work it is designed to do.

The actual work which a fan may accomplish must depend not only on its proportions, but upon the conditions of its operation and the resistances which are to be overcome. Evidently, it is improper to compare fans when operating under such conditions that these resistances cannot be definitely determined. The simplest and most natural condition of operation is that in which the fan is operated without other resistance than that of the case; that is, with open inlet and outlet. For proper comparison of different fans, the areas through which the air is discharged should bear some constant relation to the dimensions of the wheels themselves.

It has been determined experimentally that a peripheral discharge fan, if enclosed in a case, has the ability, if driven at a certain speed, to maintain the pressure corresponding to its tip velocity over an effective area which is usually denominated the "square inches of blast." This area is the limit of its capacity

to maintain the given pressure. If it be increased the pressure will be reduced, but if decreased the pressure will remain the same. As fan housings are usually constructed, this area is considerably less than that of either the regular inlet or outlet. It, therefore, becomes necessary, in comparing fans upon this basis, to provide either the inlet or the outlet with a special temporary orifice of the requisite area and proper shape, and make proper correction for the contracted vein. The fan is thus, in a sense, placed in a condition of restriction of discharge, which it approaches in practice only in so far as the resistances of pipes, passages and material through which the air must pass have the effect of reducing the free inlet or outlet of the fan.

The square inches of blast, or, as it may be termed, the capacity area of a cased fan, may be approximately expressed by the empirical formula:—

$$\text{Capacity area} = \frac{D\,W}{X}$$

In which $D =$ diameter of fan wheel, in inches.
$W =$ width of fan wheel at circumference, in inches.
$x =$ a constant, dependent upon the type of fan and casing.

An approximate value of x for general practice is not far from 3, but this is to be used only to determine the capacity area over which the given pressure may be maintained. This is not a measure of the area of the casing outlet, which is always larger than the square inches of blast. As a consequence, the pressure is lower and the volume discharged is somewhat greater than would result through an outlet having the square inches of blast for its area. But the maximum pressure may be realized when the sum of the resistances is equivalent to a reduction of effective outlet area to that of the square inches of blast.

Both the volume and the power required will evidently increase with the area of the outlet, being greater with the normal outlet than with that representing the capacity area. But this increase will not be proportional to the area; for the greatest delivery of air and the largest consumption of power will occur when the casing is entirely removed and the fan left to discharge entirely around its periphery.

Suppose it be required to design a fan to deliver 10,000 cubic feet of air per minute, through an opening of 288 square inches, which should correspond to the capacity area. Taking the value of x as 3, the equation becomes:

$$\text{Capacity area} = \frac{D\,W}{3} \text{ or } W = \frac{288 \times 3}{D}$$

Evidently it is impossible to determine the values of D and W without some knowledge of the velocity of discharge or the pressure produced.

Under the above conditions the velocity through the opening would be $10000 \div \frac{288}{144} = 5000$ feet per minute. If reference is made to tables of velocity and pressure, such as are presented in the Sturtevant Blower Catalogues, it will be found that this velocity corresponds to about $\frac{15}{16}$ onnce pressure per square inch. That is to say, a pressure of $\frac{15}{16}$ ounce maintained within a closed vessel would cause a flow therefrom through any opening at a velocity of about 5,000 feet per minute. If this pressure be maintained by means of a fan within the vessel or fan case, it will be necessary for the fan wheel to run at a tip speed equal to the velocity of discharge through the opening. Therefore in assuming a trial value for D, it is necessary to decide whether the wheel shall be small and operated at high speed or large and operated at low speed, bearing in mind that the less the diameter the greater must be the width to get the required capacity. For illustration, let us assume 24 inches as the value of D. This would require a rotative speed of $\frac{5000}{\frac{24}{12} \times 3.14} = 796$ revolutions per minute, and the width necessary to provide the required capacity area would be:

$$W = \frac{288 \times 3}{24} = 36 \text{ inches.}$$

That is to say, the wheel would be 24 inches in diameter by 36 wide at the circumference, while the case would have to be wider still. Even to those but little acquainted with fan design such proportions would appear wrong, for the width is excessive.

Suppose now we assume a diameter of 48 inches, this would call for a rotative speed of $\frac{5000}{\frac{48}{12} \times 3.14} = 398$ revolutions per minute, a fair speed for such a fan, and the width would require to be:

$$W = \frac{288 \times 3}{48} = 18 \text{ inches.}$$

This is close to the proportions of modern practice, and would be conducive to high fan efficiency.

All this pertains to operation under the condition of an outlet having an area equal to the capacity area. But in practice the casing outlet is, as already stated, considerably larger than that corresponding to the capacity area, and the working conditions may change. When resistances are considerable, they are equivalent, in effect, to a reduction in inlet or outlet area, and must be known if accurate calculations are to be made.

Although the theoretical considerations which govern the design of fans have here been given, the conditions which exist in any given case must enter into any decision as to the practical dimensions to be given the fan.

If volume alone, regardless of pressure, is the requisite, the larger the fan the less the power required. There is a strong temptation, however, for a purchaser to buy a smaller fan and run it at a higher speed; for he sees only the first cost and does not realize the entailed expenditure for extra power. If possible, a fan should never be made so small that it is necessary to run it above the required pressure in order to deliver the necessary volume. To double the volume under such conditions requires eight times the power; three times the volume demands twenty-seven times the power.

For certain purposes, such as the blowing of cupola furnaces, a comparatively small volume of air is required, but under high pressure. For exhausting, blowing boiler fires, and the like, the volume required is greater and the pressure relatively less. The former wheel requires to be narrow at the circumference, thus providing for the escape of only a small amount of air. When a fan is employed for exhausting hot air or gases, the speed required to maintain a given pressure difference is evidently greater than that necessary when cold air is handled, the difference being due, and inversely proportional, to the absolute temperature.

* * *

STUDS VS. TAP BOLTS.

The other day three mechanical men were discussing why studs or standing bolts are preferred to tap bolts for holding machine parts together, except where the studs would interfere through lack of room. One man said that studs were preferable because they were always in place and were not liable to become lost like tap bolts. The second man said this reason did not hold, because the nuts would be as liable to become lost in one case as the bolts in the other. He said that when assembling the parts of a machine studs serve to locate them and there is no fishing for bolt holes nor futile attempts to catch threads that are half covered by the edge of the upper casting. The third man, while admitting the convenience of studs in many places, said that there was another and a more important reason —a constructive one—for the preference in the use of studs.

A tap bolt, he explained, reaches through the top casting and screws into the lower one. There is, therefore, a neck of greater or less length between the head and the point where the threads engage, giving every opportunity for spring, which acts in a way to cause the bolt to work loose after it has been tightened up. That is to say, as a bolt is tightened, the threads bear harder and harder and offer a resistance to the action of the wrench, causing the body of the bolt to twist or spring. At the same time, also, the head begins to bear and the friction between it and the casting holds the bolt in the twisted condition that the wrench leaves it in. Immediately upon any jarring of the machine, therefore, this initial twist comes out, owing to the slipping of the head, and the bolt consequently works loose. With the stud, on the other hand, the bearing on the thread comes in the nut, and it is the nut that bears on the upper casting, taking the place of the bolt head of the tap bolt in this respect. Hence, in the case of the stud, these two resistances come at practically the same point; that is ,at the upper end of the stud, so that any initial strain which may exist in the body of the stud can have no tendency to loosen the nut.

All three men agreed that there might be reason in this and accepted it as the true explanation of the question, at least until some one should come along who could give a better one.

AMONG THE SHOPS.

PRACTICAL IDEAS FROM OHIO, GATHERED AT CLEVELAND AND SPRINGFIELD.

AT CLEVELAND, OHIO.

Fig. 1 shows a Pratt & Whitney machine with a special foot stock attachment for milling the circular slots in the trimmer tables, as made at Bardons & Oliver, of Cleveland, O. The foot stock is moved out so as to show the slot more clearly, and it will be noticed that the work rests on a piece, B, fastened to the platen of the machine, with a flange to gauge the position of the work and provide a place for starting into the work. The cutter swings under and out of the way while changing from the slot in one end of the table to the other. The slots are of a dovetail shape and the construction of the foot stock is so apparent that further description would be needless.

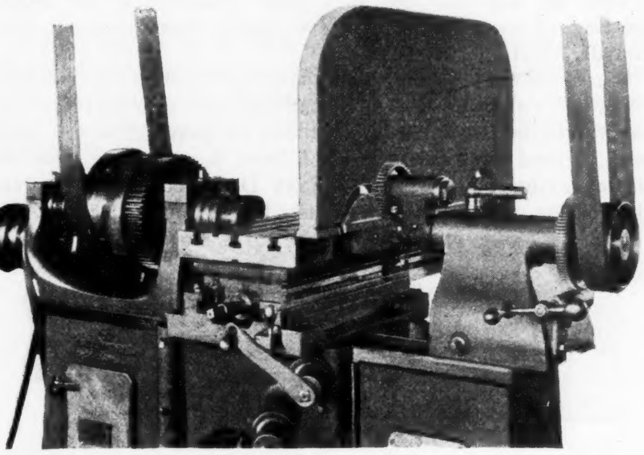


FIG. 1. MILLING A CIRCULAR SLOT.

Bardons & Oliver very kindly furnished a drawing of the undercut planer tool, Fig. 2, but the photograph seems to show the construction of the tool sufficiently. The piece, C, is pivoted so that it can be turned and clamped firmly by a nut on the back side of the shank, D, with the tool, E, in any position. The tool block, F, works on a pivot exactly like the clapper in a planer head and, of course, its position makes the spring, G, necessary.

Fig. 3 is another specimen from the same shop and represents a very convenient form of cutter head, with facing tools in position. It is made in two parts so that it can be readily removed while the boring bar is held in position at both ends. It is only necessary to loosen the set screw and slip the block, H, out to one side along the bar and the other part will drop off. Yet the whole is locked firmly when the set screw is tightened

FIG. 2.

BROWN HOISTING AND CONVEYING MACHINERY CO.

The visitors' register at the Brown Hoisting and Conveying Machinery Co., Cleveland, O , was the first thing to attract my attention on entering the office, and I am surprised that more shops do not have them. But it is about as unusual to see such a generous supply of hoists and cranes about a plant, and the odd design of the traveling crane operating in the main shop makes it interesting. In the yard is another type, supported in the middle, and extending across the railroad and wagon entrances, with a capacity for raising five tons and placing it anywhere on the line of some 300 feet. Five tons on the end of a 150-ft. lever is a pretty good load, but I state the figures as given me by one who is in a position to know.

My brief visit late in the afternoon permitted only a superficial view of the different departments and I cannot go into details; but one useful article worth mentioning had several men around it laying out work. It was a strongly built surface plate about 9x12 feet. Of course it was not scraped to the extreme accuracy of the common little tool room affairs, but is far more useful for large work and is undoubtedly true enough for its purpose.

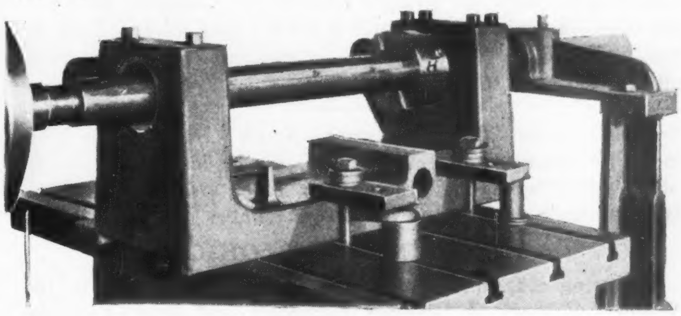
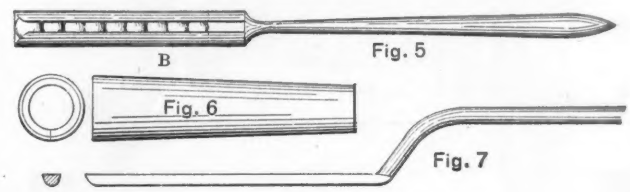


FIG 3. A CONVENIENT CUTTER HEAD.

WHITE SEWING MACHINE CO.

Presuming that Mr. Brady will include "The White" in his series of papers on "Sewing Machinery," I will omit it; but bicycles and bicycle pedals have become quite an important branch of the business done by the White Sewing Machine Company, and very likely the little tools illustrated in the sketches herewith have escaped previous attention. Fig. 5 is a simple and practical tool for loading bicycle bearings, and is shown in addition to the one previously mentioned, as a matter of variety. B is a piece of thin brass tube, slit on one side for observation, and soldered to a handle. B holds just enough balls for a bearing, and is readily filled by a scoop through a box full of balls.

Fig. 6 will be recognized as the part of a bicycle pedal shielding the pedal pin. It is made from sheet metal, and has a brazed joint on one side, and the brazing material is distributed very evenly along this joint on the inside by a tool like Fig. 7, which



is made from a piece of wire filed flat on top and of the right width to pick up just the right amount. By dipping this into the brazing material a small quantity is taken up like so much sugar, and when it has been inserted into the piece to be brazed a turn of the wrist deposits it on the seam.

WARNER & SWASEY.

What pleased me most at Warner & Swasey's was to find them doing even a finer class of work than I expected Besides the regular machine tools built by them the graduating engine used in making telescopes, range finders, etc., is a product of their own shop. Mr. Warner explained that the error in the dividing wheel did not exceed one inch in three miles, as the most comprehensive way of expressing one second of an arc.

To me these were among the most interesting shops in Cleveland, but in this case details from another source have been suggested and may even precede these notes.

Before leaving I examined the first 20 pages of their visitors' register, and found prominent names too numerous to mention. Even the different foreign cities represented cannot be given here, so I will only name the fourteen foreign countries to be noticed: China, Canada, Sweden, England, Japan, Germany, Syria, Scotland, Austria, Switzerland, France, Holland, Belgium and Hungary.

Every name on those twenty pages represents a visitor that has been welcomed to a shop that is sufficiently interesting to attract visitors from all over the world, and I recall the recent items by "Milo" and "Been There" on the subject of "No Admittance," which caused the writer to count the no-admittance signs while going through a certain shop in the Empire

State. There was a grist of them that would cost more for painting alone than a first class register made to order.

Coated with dirt like the gloomy interiors of the several departments of this same institution, I read this supplementary notice:

"Workmen must pay for all broken lamps."

In speaking of this no admittance subject to a certain engine builder, I referred to Prof. Sweet's policy of welcoming visitors, and he said: "Well, the Straight Line Engine Company have got the engine subject down to such a science they have nothing to fear." Those few words are full of meaning, and are a compliment too good to keep.

A visit to the Cleveland Machine Screw Company was even more interesting than I anticipated, but a description of my observations would be practically a repetition of Prof. Benjamin's article published in the March number.

AT SPRINGFIELD, OHIO.

In Springfield, Ohio, is a group of buildings occupied by the Warder, Bushnell & Glessner Company, and devoted to the manufacture of binders, mowers and reapers; and equaling in capacity a single structure one story high, fifty feet wide and over three miles in length. Here is an agricultural machine works employing about 1,500 hands that is interesting all the way through. Many practical devices are to be seen, and these, together with the evident success of the company, speak well of the combined mechanical and executive ability of its manager, Mr. Charles A. Bauer.

The works include two of the most attractive foundries that I have yet seen. The larger of these is a building 80 feet wide and 600 feet long, without any of the posts, gib cranes or similar obstructions so common to this department elsewhere. There are the usual railway conveniences on the floor, and a number of hoists operating on overhead tracks, which extend about half the width of the building, serve a sufficient portion of the floor for the heavier part of the work. The view through the malleable iron foundry is also pleasing. Even the paint shop should be included in a tour of the place by a stranger. But I have said in the past that it is not my purpose to consume much space with descriptions of plants, and, proceeding to the laboratory we fine among the subjects more common to its sphere one that appeals to any machinist who has had to contend with the needlessly hard casting nuisance. This is the drilling test to determine the degree of hardness, as compared with standards that have been found most suitable for their use. The testing apparatus in this case consists of a suitable upright drill, with a revolution counter attachment and electrical connections for starting the same as soon as the drill begins to cut full size. The number of revolutions required to drill just one-half inch from this point represents under specified conditions the degrees of hardness of the casting; the greater the number, the harder the casting.

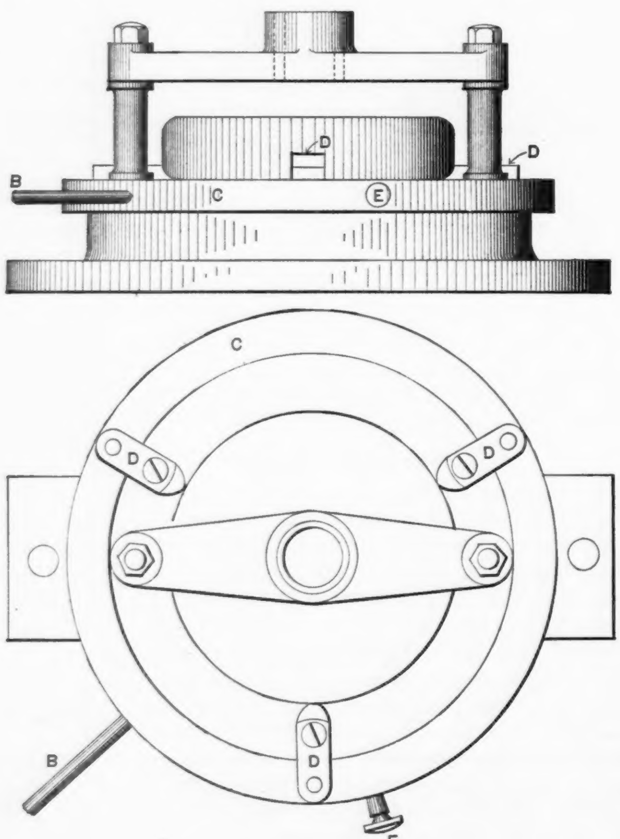


FIG. 8. A DRILL JIG.

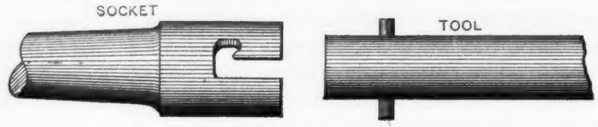


FIG. 9.

This test was described in a paper presented at the recent meeting of the American Foundryman's Association in Cincinnati, by Mr. C. L. Bauer.

Fig. 8 shows a specimen of drill jig found in the machine shop, and with its various modifications, more or less resembling those described in my notes published on page 240 of the April issue, is used quite extensively. A lever B, projecting from the movable ring C, to which the jaws D are connected, enables them to be operated simultaneously like a universal chuck, except that the work is released by a cam motion of the work end of each jaw as it swings on the pivot, when the outer end is thrown to one side of the radial position shown, by a movement of C. In this particular case the old bayonet hitch drill socket (Fig. 9) was used, the proper speed for the size of the tools being so slow that they could be readily changed without stopping the machine. E is a tapered pin for locking the ring C.

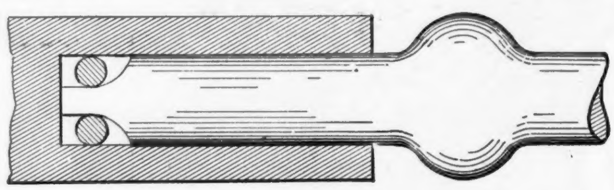


FIG. 10.

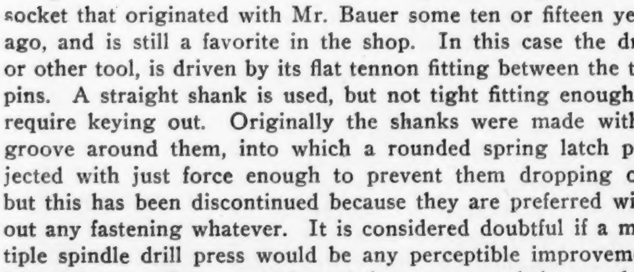


FIG. 11.

Fig. 10 illustrates another form of quick change drill and socket that originated with Mr. Bauer some ten or fifteen years ago, and is still a favorite in the shop. In this case the drill, or other tool, is driven by its flat tennon fitting between the two pins. A straight shank is used, but not tight fitting enough to require keying out. Originally the shanks were made with a groove around them, into which a rounded spring latch projected with just force enough to prevent them dropping out, but this has been discontinued because they are preferred without any fastening whatever. It is considered doubtful if a multiple spindle drill press would be any perceptible improvement in some cases where a number of these are now being used on ordinary single spindle machines. An explanation of the enlargement on the shank is needless if the reader has ever tried to push a drill into a socket when it was in motion. I believe the market is being supplied with this style of tools at the present time by at least one manufacturer. Fig. 11 shows a style of taper gage that is used here. The slit provides a sharp corner the entire length of the taper for observing the fit, and as the amount of variation can be seen at a glance before it is turned down small enough to project through the end, it seems as if this would facilitate setting the taper. Of course such a gage would have to be stiff and carefully ground after splitting, and

I am not suggesting it to those who ream a hole in the first piece of cast iron to be found and use that for a standard. Neither would I recommend it as a reference gage for the final inspection of work; but for the ordinary shop use it appears to be all right.

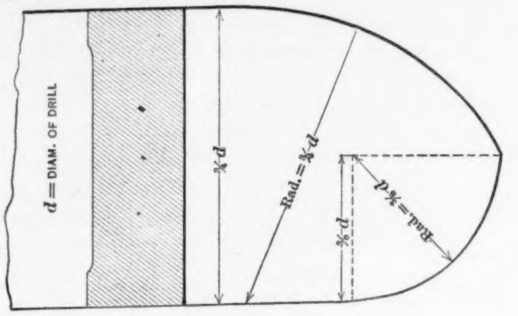


FIG. 12.

Fig. 12 represents the form of cutter used by the W. B. & G. Co. for fluting twist drills, and they produce a lip that is practically straight when ground to the usual angle of 60 degrees from the centre line of drill and the flute is milled at an angle of 20 degrees.

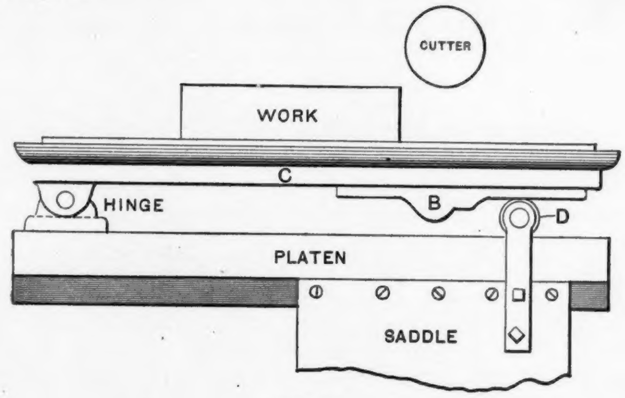


FIG. 13.

Several machines of the Lincoln type were rigged up for bridge milling, and in some cases adapted to cuts of varying depth by the use of a suitably shaped guide B, fastened to the under side of the auxiliary table C, so as to ride on the stationary bridge D, as shown in Fig. 13. By this method key ways were being milled in the middle of shafts as readily as if beginning at the end.

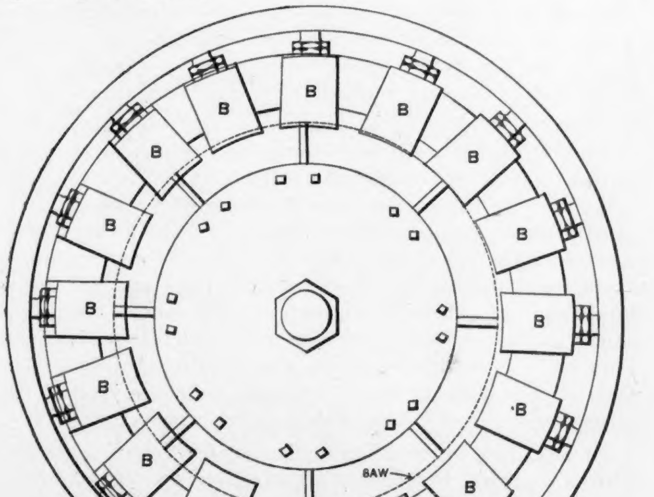


FIG. 14. MILLING MOWING MACHINE FINGERS.

The method of milling the slot in the fingers for mowing machine fingers, through which the cutters work, also interested me, and an outline sketch looking down on the top of the machine is reproduced in Fig. 14. The milling saw is made in sections, bolted between heavy flanges on the top of a vertical shaft, and a series of holders B, fitting by tongue and groove at each side, are carried around the saw by the slowly revolving carrier C, and at the same time fed towards the centre of the saw by the spiral of stationary guide D. The depth of the cut is determined by the check nut adjustment on each holder, and when a holder has made one revolution around the saw it is removed at the gap in front and another with two more of the fingers is inserted and starts on its trip around the saw with the continually revolving carrier. These holders travel in a circle about four feet in diameter, and just about keep a man busy changing the work and putting them in as fast as they come around to the opening in the guide.

No attempt has been made to work to scale or show minor details in these sketches, as they are prepared from memory; and no mention has been made of other more interesting subjects, having omitted providing myself with suitable photographs which would be the only satisfactory way of illustrating them.

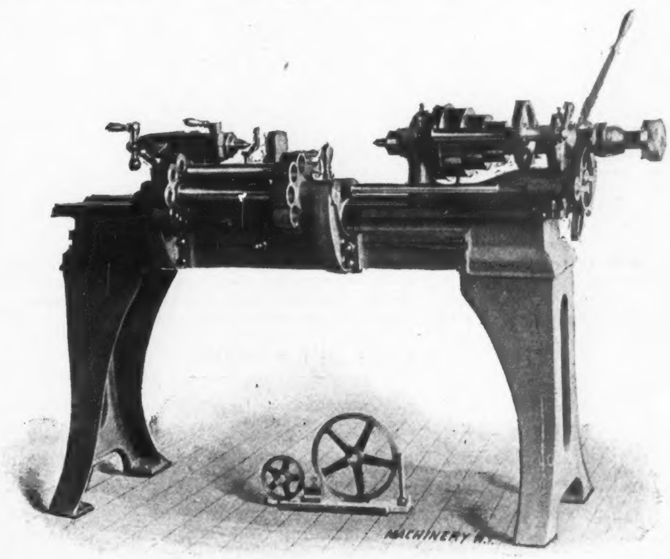
A. L. G.

* * *

A SPECIAL LATHE FOR THREADING TAPS.

We illustrate in the accompanying view a new lathe for threading special taps that possesses features of interest. The idea of the design is to have a lathe so arranged that small orders of, say, half a dozen of a kind, can be cut economically, and at the same time in the most accurate manner.

As will be noticed, there is a "turret" on the back side of the lathe carrying eight lead screws of different pitches. There are also eight half nuts to correspond with these lead screws, one of which appears on the floor in the photograph. These engage in the lead screw the same as an ordinary split nut in a regular lathe. These lead screws are 14 inches long, 1¾ inches diameter, and with the half nuts have ratchet threads so that in case the half nut is not fully engaged with the lead screw there can be no variation whatever in the work. With these eight lead screws and half nuts all the standard threads from 4 to 26 can be cut, and a tap or stud up to 12 inches in length can be made. By the use of longer lead screws, which can be supplied to order, longer work can be done just as advantageously.



A NEW SPECIAL LATHE.

It will be noticed that only two change gears are necessary, and by their use three different pitches of thread can be cut with any one lead screw, while to make a still further change it is only necessary to revolve the turret and put another half nut in place to correspond with the lead screw then in place. This can be done in a moment's time and the machine is ready for work. The carriage has a cross-feed screw, just as on an ordinary lathe, but it is used only to feed the tool into the work. It is thrown out instantly and at one motion, by the use of the hand lever on the side of the carriage, which is clearly visible at the center of the photograph.

This lathe is manufactured by the Florence Machine Company, Florence, Mass. The first one of the lathes to be built has been thoroughly tested by one of the leading maufacturers of taps and dies in the country, and has given such excellent satisfaction that others have been ordered. The lathes are made with beds 5 feet long, 1 1-16 inch hole through the wire feed, and with phosphor bronze boxes. Weight, 1,100 pounds.

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AUGUST, 1898.

A friend in San Francisco writes that a large package of magazines and other reading matter was recently made up in that city to send to Admiral Dewey's men at Manila, and that in the package were some copies of MACHINERY. These men are many of them good mechanics as well as able fighters, and we feel sure that this action of some thoughtful person will be appreciated.

* * *

Any man, in starting in to design a new line of machines, who will first decide upon a series of pins of various sizes, the fewer of them the better, and then will design his machines as far as possible to use these pins, and these only, will win everlasting gratitude from those who follow in his footsteps, or from those who have anything to do with the manufacture of the machines. As a case in point, a man who undertook to systematize the methods of manufacture in a printing press factory found that on one size of press there were 29 different sizes of pins that were used for rolls and other purposes, and that in designing other sizes of presses no effort had been made to work in any of these first 29 pins.

In order to keep down the costs in machine construction the pins must be made in the screw machine, and to be made economically in the screw machine they must be made in quantities. To carry a large stock of pins, however, where only one of a kind, or at most only a few of a kind, are used on a machine, and on only one machine of a series, costs a great deal of money—much more, in fact, than at first sight seems possible without figuring it up in detail. This is only one of the many practical points that a draftsman should take account of, and the ones that are able to look out for these things are the ones that are sooner or later sure of the best positions.

WRITTEN REPORTS IN THE SHOP.

In any well-conducted business it is customary to have statements drawn off at intervals to show by direct comparison and in small space the relative expenditures and receipts of the past and present and any other information that may assist in arriving at the condition of the business or in shaping its future course.

In the shop the management is not directly concerned with financial matters to so great an extent as the office, but there seems to be no reason why there should not be a drawing up of accounts, as it were, at regular intervals, to assist those who are expected to keep the plant running on a paying basis and to enable them to keep in closer touch with its operations.

We have reference to a system of written reports made out at intervals by the heads of departments, inspectors and others. The danger in introducing such a system is that with it will come more red tape, which at best is more or less of a bugbear in every shop. In spite of this, however, we believe that the tendency in shop management is to let things go at loose ends, to trust to word of mouth rather than written order, and to have no effective means for comparing past and present results other than that afforded by the cost system. Our attention was first called to this fact at one of the large shops of the country which was for a time under the management of one man, who acted both as general manager and superintendent. Naturally his time for the consideration of any one subject was limited, and he adopted the plan of having reports sent to him daily from the different departments to assist him in his management. Since then several incidents of the adoption of a system of shop reports have come to our notice, which have given beneficial results in several directions.

In the first place, such a system serves to place before the superintendent every morning, or at stated intervals, information about the standing of the work that he would himself gather into his note book, mental or otherwise, during his rounds through the works. We do not advocate having these reports take the place of a tour of the works, which would take away the superintendent's best means for keeping the wheels in motion; but it would be of assistance to look them over before making the rounds, and they would also give a permanent record for future reference, in case of delay or trouble.

Such a system makes it very convenient for foremen or others to place on file requisitions for supplies or tools. It also serves as a check on the use of supplies that are not included in the regular stock-room reports, especially in the engine room, where the coal, water and supplies should be included. The reports can, in fact, be made to include daily readings of all water and gas meters, if these items represent a sufficient expenditure.

Another case in point is the inspection of elevators, shafting, belting, boilers, cranes and other machinery. It is a good thing to know what has been inspected, when and by whom, and to have a signed report made out by the person making the inspection. The same thing holds true, also, in making repairs. Pumps and engines that need packing, traps and heaters that need cleaning, and machines that are continually breaking down would all be recorded at headquarters, with full details, where a fund of information would soon accumulate that would be of material assistance in buying supplies and in rendering judgment concerning the value of this or that appliance.

These are but a few hints of the possibilities of a system of written reports. They are in principle but an extension of the time and stock room reports that are in use in every shop. In adopting the system blanks should be provided, which, when filled out, will give the superintendent the information regarding the details of the plant in concise form, so that he can grasp them in a few minutes and act accordingly. The mere fact that the superintendent is attending to these points in a methodical and careful manner will not only impel his subordinates to do the same, but will stimulate them to increased interest in these matters and make them quite as anxious as anybody to keep watch of them. A report, moreover, serves to shift the responsibility in matters where a person has no control to the person who has that control, and thus tends to hasten the execution of needed orders.

WHAT A MACHINE DESIGNER SHOULD KNOW ABOUT SPRINGS.—Concluded.

J. Begtrup.

Torsion Springs.

What really happens to the molecules of a bar when it is twisted within the elastic limit is a matter of conjecture, but all formulas for strength and deflection of torsion springs are based on the assumption that the molecules receive a sort of lateral or sliding displacement, as if subjected to a shearing action. Whether or not this assumption is correct it is certainly supported by experimental results. It is, for instance, known that the angle of deflection is directly proportional to the twisting force, which fact would hardly agree with other theories.

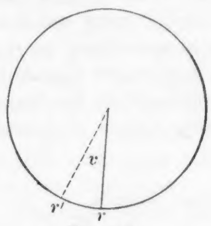


Fig. 15

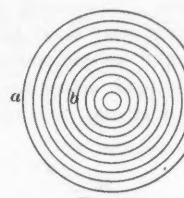


Fig. 14

Fig. 14 represents a cross section of a steel rod divided into a number of imaginary concentric rings of equal thickness. The torsional strength of this rod depends on the resistance to shearing of the rings and on their respective distances from the center, which are their leverages of resistance. Ring a is twice as large as ring b and is twice as far removed from the center, and offers, therefore, $2 \times 2 = 2^2 = 4$ times the resistance to a twisting force. Suppose we have another rod twice as large in diameter and divided it in the same number of rings, then each ring will be twice as thick, twice as long and twice as far removed from the center as the corresponding ring of the first rod; the torsional resistance will therefore be $2 \times 2 \times 2 = 2^3 = 8$ times that of the first rod, provided the resistance per unit area is the same in both cases. In other words, by increasing the diameter of the rod we increase both the thickness, length and leverage of resistance of the rings in the same proportion. An increase of diameter has, therefore, a triple effect on the torsional strength, which, therefore, becomes proportional to the third power of the diameter.

The formulas for torsional strength are

$$\text{For round bars } R\,W = \frac{\pi}{16} Z\,d^3 = \tfrac{1}{5} Z\,d^3 \text{ nearly} \dots \quad 1$$

$$\text{For square bars } R\,W = \frac{1}{3\sqrt{2}} Z\,d^3 = \tfrac{1}{4} Z\,d^3 \text{ nearly} \dots \quad 2$$

where W = twisting force in pounds.
R = lever-arm in inches.
Z = shearing unit stress of outside ring, in pounds.
d = diameter of bar, in inches.

For tempered steel we may put Z = 80,000 pounds, and the moment of resistance of round steel = 1-5, 80,000 d^3 = 16,000 d^3. This gives for a ½-inch rod safe moment of resistance = $(\tfrac{1}{2})^3 \times 16{,}000 = 2{,}000$. If we twist the rod with a 6-inch lever the safe load on the end of the lever = $\frac{2000}{6} = 333$ pounds. A ⅝ rod would carry $\frac{16000}{6} \times (\tfrac{5}{8})^3 = 650$ pounds on the end of a 6-inch lever. It will be noticed that a small increase of diameter greatly increases the strength, and that square steel will carry about one-fourth more than round steel of same diameter.

We will now consider the torsional deflection. Fig. 15 is an end view or section of a twisted steel rod, r and r^1 are imaginary radial lines, and r is supposed to be in a plane above r^1 and is supposed to have just covered r^1 before the rod was twisted, that is, a small particle directly over r^1 is moved horizontally a distance r^1r through an angle v. Fig. 16 is an elevation of part of the rod where the dotted lines indicate the twisting of the surface much exaggerated. Plans r and r^1 are supposed to be 1 inch apart and rp represents the transverse displacement of a small particle originally at p. The maximum unit stress in each transverse section of the rod is supposed to be equal to the product of this displacement and a certain constant multiplier. If the material be tempered cast steel and Z = 80,000, the distance pr is about 1-150 inch. It varies directly as Z and is independent of the diameter of the rod. The multiplier is in this case 12,000,000, which, according to our hypothesis, is a constant for tempered steel. It is a purely hypothetical quantity, which bears no rational relation to the modulus of elasticity of the material, but we may call it the torsional modulus of elasticity, because it takes the same place in the calculation of torsional deflection as the modulus of elasticity takes in the calculation of bending deflection. It will be seen that a rectangular area on the surface of the rod becomes a rhomboid when the rod is twisted. Area rr^1ab is a rhomboid or deformed rectangle, and supposing pr^1 to represent unit of length, and let distance p r be the displacement caused by a torsional unit stress of one pound at the surface of the rod, then this displacement becomes the modulus or measure of deformation, and which is the reciprocal of the modulus of electricity; but it will be readily inferred that such deformation does not produce a lateral or shearing stress, as if the surface had been stretched lengthwise of the rod a distance equal to pr, and that the torsional modulus of elasticity must be considerably less than the modul of elasticity for bending. We have seen that for a given maximum unit stress Z the moment of torsional resistance varies as the third power of the diameter; but without this limitation of stress the mean unit stress for any given angular deflection varies directly as the diameter of the rod, and under this condition an increase of the diameter has a quadruple effect on the moment of resistance, which, therefore, becomes proportional to the fourth power of the diameter; and the deflection will be inversely proportional to this. That the entire angle of deflection must be proportional to the length of the rod requires no demonstration. It is also directly proportional to the load.

The following are convenient formulas for torsional deflection:

For round sheet

$$F = \frac{32\,W R^2 l}{\pi\,G\,d^4} = \frac{10\,W R^2 l}{G\,d^4} \text{ nearly} \dots \quad 3$$

$$F = \frac{2\,Z\,l\,R}{G\,d} \dots \quad 4$$

For square sheet

$$F = \frac{6\,W R^2 l}{G\,d^4} \dots \quad 5$$

$$F = \frac{\sqrt{2}\,Z\,l\,R}{G\,d} \dots \quad 6$$

F = linear deflection at end of lever.
W = twisting force at end of lever.
R = length of lever.
l = length of rod.
G = torsional modulus of elasticity.
Z = unit shearing-stress in periphery of cross section.
d = diameter of rod.

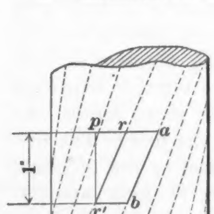


Fig. 16

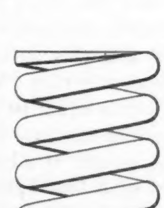


Fig. 18

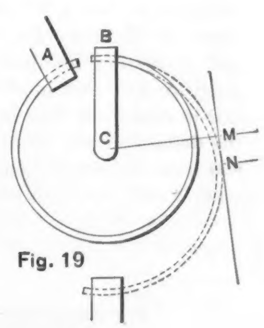


Fig. 19

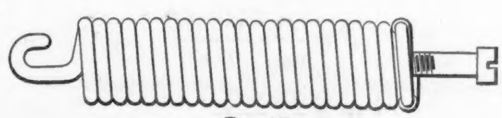


Fig. 17

For spring steel G = 12,000,000 appears to be a nearly correct mean value, according to my own experience. The proper value of Z depends on the working conditions. A spring that is continually working should be strained less than one whose action is intermittent or irregular; and it should be observed that shearing resistance at the elastic limit is somewhat less than tensil strength at the same limit. Z = 80,000 is probably not too much, unless the spring is continually working to its full capacity. But when the construction and circumstances are so as to admit of a lower stress it is always preferable. As a simple example of torsion springs, take a rod of ½-inch round cast steel 3 feet long, fixed solidly at one end and the other end so guided as to

prevent lateral motion, and let there be a 6-inch lever keyed to this end. How much will it be safe to load the end of the lever if the rod is twisted 100 times a minute? The rod is not supposed to be tempered, and though its ultimate strength is considerable, the elastic limit is comparatively low. Let Z = 30,000 and E = 12,000,000, then in substituting in formula 1 we have

$$6\ W = \tfrac{1}{5}\ 30{,}000\ (\tfrac{1}{2})^3 = \tfrac{6000}{8} \text{ and } W = \tfrac{1000}{8} = 125$$

pounds, which is the admissible force on end of lever. The deflection for this force can easily be found from formula 4, because the value of Z is known. We have

$$F = \frac{2.30{,}000.36.6.2}{12{,}000{,}000} = 2\tfrac{1}{16} \text{ inch.}$$

If we had this rod of tempered steel we might put Z = 70,000 and would then have $W = 125 \times \frac{7}{3} = 292$ pounds and $F = 2\frac{1}{16} \times \frac{7}{3} = 4\frac{13}{16}$ inches.

Steel used for springs should have a high elastic limit and preferably a low modulus of elasticity, for the deflection is proportional to the quotient $\frac{Z}{G}$ and the greater efficiency of torsion springs is due to the smaller modulus of elasticity, as compared with that of bending. For the same unit stress at the surface of the rod the angular deflection will vary inversely as the diameter, which is an important rule easy to remember. But for the same load and varying diameters the deflection varies inversely as the fourth power of the diameter. The torsional deflection of a ⅝-inch rod, for instance, would only be about 2-5 of that of a ½-inch rod under the same load.

The rod would in many cases have to be very long to give the desired deflection, and a straight rod would therefore often be impracticable; but fortunately it can be bent so as to make a comparatively short spring easy to make and easy to temper. This is obtained by bending it in the form of a cylindrical helix, or screwline, as shown in Figs. 17 and 18. One of these springs will be compressed and the other will be stretched, but the former may by a slight change in the connections be used both ways. These are true torsion springs, though it may not appear so by first sight. The following analogous case will explain it. Fig. 19 shows an open ring of steel wire firmly fixed and supported at A, and a radial lever firmly attached to the free end at B. A pressure exerted on this lever at the center of the ring perpendicular to its plane will twist the wire while it pushes point B back. This will be better understood by reference to the bent wire, shown in dotted line. At a point N is drawn a tangent and from C a perpendicular CM. There will be a bending moment at N represented by line MN and a twisting moment represented by line CM; but when the curve becomes a circle with center at C the bending moment disappears and there is nothing but a twisting moment left, and this twisting moment is constant for any part of the concentric ring. We see that when the rod is coiled the twisting lever is equal to the mean radius, and the deflection will be in line with the axis of the helix. The helical form is compact and the weight of a helical spring of round steel is only about $\frac{5}{12}$ of that of a leaf spring of same capacity.

For round steel

$$W = \frac{40}{100} \cdot \frac{Z\,d^3}{D-d} \quad \ldots\ldots\ldots\ldots\ 7$$

$$F = \frac{8\ W(D-d)^3}{G-d^4} \quad \ldots\ldots\ldots\ 8$$

$$F = \frac{314}{100}\ \frac{Z(D-d)^2}{G\,d} \quad \ldots\ldots\ldots\ 9$$

For square steel

$$W = \frac{47}{100}\ \frac{Z\,d^3}{D-d} \quad \ldots\ldots\ldots\ldots\ 10$$

$$F = \frac{47}{10}\ \frac{W(D-d)^3}{G\,d^4} \quad \ldots\ldots\ldots\ 11$$

$$F = \frac{222}{100}\ \frac{Z(D-d)^2}{G\,d} \quad \ldots\ldots\ldots\ 12$$

Dr. Reuleaux has called attention to this in his first work on springs, which was published in 1857, and in his latest work he proposes the use of helical springs for railroad cars; from which we may infer that such springs are not common on German roads, while they have certainly now become very common in this country. Formulas 1 to 6 may be used for the calculation of helical springs, but are more convenient.

In these formulas F is the deflection of one coil and D is the outside diameter of coil, and the meaning of the other letters is the same as in formulas 1 to 6. It appears from these formulas that square steel is about 17 per cent. stronger than round steel, but for the same unit stress the deflection of square steel is about 30 per cent. less. Round steel is, therefore, better adapted to torsion springs. This may easily be perceived without any calculation, considering that when square steel is twisted the corners cannot add very much to the strength on account of the smallness of their areas, which terminate in four points; but these points being furthest removed from the center will take the greatest strain and will limit the angle of deflection as much as a full circle, including the points, would do.

Fig. 18 shows a car spring of 1-inch round steel, 5 inches outside diameter. How much will it carry? It must not close under the maximum static load, but it may close entirely by the jolting of the car, and we will therefore put Z = 50,000 pounds for maximum static load, assuming the elastic limit to be above 100,000 pounds unit stress. Substituting these values in formula 7 we have:

$$W = \frac{40.\ 50{,}000}{100.4} = 5000 \text{ pounds.}$$

and assuming Z = 100,000 pounds when the spring is entirely closed we have from formula 9:

$$F = \frac{314.\ 100{,}000.16}{100.\ 12{,}000{,}000} = \tfrac{7}{16} \text{ inch.}$$

That is, the coils should be 7-16 inch apart without load, and they will be 7-32 inch apart under maximum load. The spring shown in Fig. 17 is 3 inches in diameter on the outside and is made of ½-inch round steel, and there are 24 coils. How much may this spring be extended if used on a shaft governor? As its work is intermittent, and as it very seldom is fully extended, we may put Z = 70,000, and we have from 9:

$$F = \frac{314.\ 70{,}000\ (2\tfrac{1}{2})\ 2}{100.\ 12{,}000{,}000.\tfrac{1}{2}} = .23 \text{ inch,}$$

which is the allowable deflection of one coil, and .23×24=5½ inches is, therefore, the safe extension of this spring. From formula 7 we find the maximum load to be 1,375 pounds. I have calculated a table of carrying capacity and deflection of helical springs of round steel varying from No. 16 to 1 inch diameter. It was first published in the American Machinist, and may now be found in Kent's Pocket Book, or, if preferred, blue prints from the original will be furnished by me. Close coil springs, as represented by Fig. 17, are sometimes distinguished by a considerable initial tension; that is, it takes some initial force to separate the coils, and the elongation cannot be calculated from the above formulas. The probabilities are that they are made from cold rolled wire, untempered, for the initial tension would be removed by the process of tempering. Such springs are easily distinguishable by their resistance to bending before they are stretched.

It will be noticed that in my calculations of springs the supposed elastic limit is approached closer than would be judicious in the calculation of other machine parts; but my results agree with the average common practice, and there are several reasons why this is so. In the first place, springs are made of cast steel of moderate dimensions, which is the most reliable material known. In the second place, the form is so that no part can be subjected to unexpected or unaccountable strains, and on account of their great elasticity they do not suffer by shocks or blows. Lastly, springs are not used where their failing would be of any serious consequence.

There seems to be considerable uncertainty or lack of knowledge as to the proper modulus of elasticity of tempered steel. The comparatively small demand for such knowledge except for the calculation of springs is a probable reason for its scarcity. According to various tests the modulus of elasticity of untempered steel is from 28,000,000 to 32,000,000, and it appears from calculations of bending and twisting deflection of ordinary springs that the modulus of elasticity is not increased by tempering. Still it will hardly do to overlook the figures given by Dr. Reuleaux, which appear in his "Constructeur" of 1872, and are repeated in the enlarged edition of 1882-1889. His figures for the elastic limit and ultimate tensile strength are also interesting, and I give them here in English units. In the heading he

states that the figures are mean values of numerous experiments by various experimenters on materials of different make, and in actual use. He gives only round figures, in milimeter-kilogram units, and I give the exact translation in inch-pounds:

	Modulus of elasticity.	Elastic limit.	Ultimate tensile strength.
Spring Steel, tempered	28,440,000	71,000 to 99,500	113,700
Cast steel, untempered	28,440,000	35,500	113,700
Cast steel, spring tempered	42,600,000	92,000 to 213,000	142,000

I should add that the formulas which I have given for coiled springs are not new. They were first worked out by Dr. F. Reuleaux, about forty years ago, and they may be found in a his "Constructeur."

I have made some effort to learn the characteristics of so-called "spring steel," but the results of my inquiries seem to be of a rather vague or negative nature, though the great difference in elasticity and strength, as given by Dr. Reuleaux, would lead us to expect a marked distinction in composition or structure. The following letter from an expert on crucible steel is of interest in this connection:

Mr. J. Begtrup.—Dear Sir: Replying to your favor of December 16, I know of no "peculiarities of composition" of spring steel except that spring makers have made their prices so low that they are using the cheapest steel they can buy in order to save themselves.

The same rule holds good in springs as in almost anything else—the better the steel the better the spring; but if a cheap spring will wear out a car or a locomotive, what is the use in making a more costly one? A good spring can be made of steel of any carbon from .60 to 1.30 by proper hardening and tempering. The mildest steel is hardened in water, the medium in oil and water and the hardest in oil. All will be good if they are worked properly. Yours truly, WM, METCALF.

* * *

A NEAT CUTTER ARBOR.

The inclosed photograph and sketch needs but little explanation, as they clearly show means for removing stem cutters from their arbors by a wrench, rather than the common—and frequently disastrous—one of striking upon the teeth. Fig. 1 shows

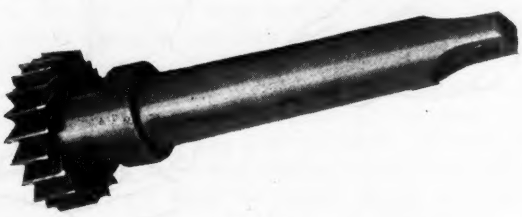


FIG I

the arbor and cutter as they appear, with the cutter screwed in place, and Fig. 2 shows the details of construction. There is a collar back of the cutter, with two flats milled, as shown, to fit a wrench; and the collar is also tongued to fit a groove in the

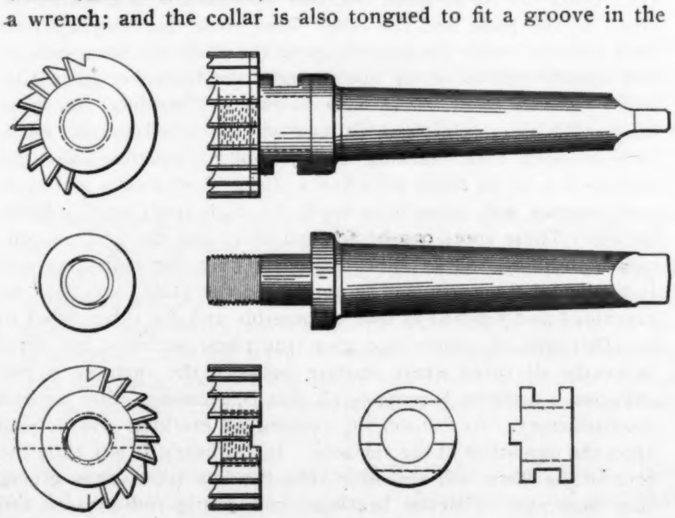


FIG 2.

cutter. To remove the cutter from the arbor, therefore, all that is necessasry is to give the collar a slight turn with a wrench, which will start the cutter and enable it to be removed by hand.

Grand Rapids, Mich. E. L. O. K.

SHOP TALKS WITH YOUNG MECHANICS.—6.

SCRAPERS AND SURFACE PLATES.

W. H. VAN DERVOORT.

The scraper is a tool used by machinists for producing truer surfaces than can be produced by the ordinary planing and filing processes. It is strictly a tool to be used on stationary work, although the distinction between it and the hand turning tool used by the brass worker is not clearly drawn.

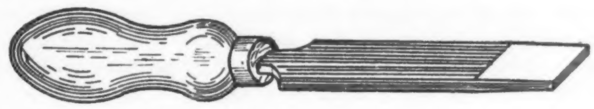


FIG. 99.

The flat hand scraper, as usually formed, is shown in Fig. 99. It is forged from a piece of flat steel of from ¾ to 1¼ inch in width by ⅛ to 3-16 of an inch in thickness. The point is drawn down so that the end is about 1-16 of an inch thick, as shown in Fig. 100. The flats should be ground well back from the point and the end at right angles to the length of the tool, as shown at A, Fig. 100, thus making the angle of the cutting edges but slightly more than 90°, as shown at B, same figure.

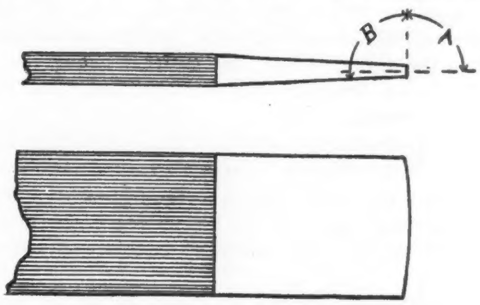


FIG. 100.

The end should be ground slightly rounding in its length to prevent the corners from digging into the work and the tools taking too broad a cut, which tends to produce a wound or chattered surface.

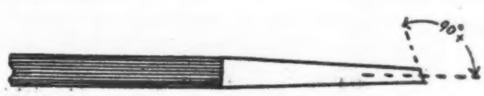


FIG. 101.

If the end is ground so as to give one side a keener cutting edge, as shown in Fig. 101, this edge will cut faster, but the surface produced will ordinarily not be so smooth as in the former case, it being difficult to prevent the tools chattering.

The scraper, including handle, should be from 10 to 12 inches long, depending on the size of stock and the character of the work on which it is to be used. If too long it will be springy and will not do good work. As the angle forming the cutting edge must be kept very sharp, a high temper is necessary, and the end faces after being ground must be oil-stoned often in order to make the tool cut properly.

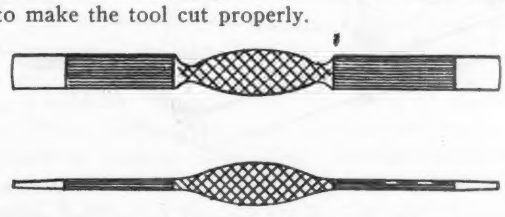


FIG. 102.

The double ended scraper shown in Fig. 102 is a form frequently used. This scraper should be made somewhat longer than the one shown in Fig. 99, from 14 to 16 inches being about right. The central portion, which serves as a handle, should be enlarged and knurled, or twisted in the forging, so as to enable the hand to grip it firmly.

A form of scraper shown in Fig. 103 is sometimes employed on fine work. The disadvantages of this form arises from its hidden cutting edge while at work, and its having but one cutting edge, thus necessitating more frequent grindings than with the straight tool.

The scrapers shown above are suitable for use on plane or

convex surfaces. If a concave surface is to be worked upon a scraper of semicircular cross section, as shown in Fig 104, will be used.

Frequently in scraping circular surfaces, and more especially in the softer metals, as brass or babbitt, a three-cornered scraper can be used to advantage. Such a tool is shown in Fig. 105. The cutting edges are long ones, formed by the intersection of the sides. In its use this tool is held in both hands, by the point and handle, when the nature of the work will permit, the cutting edges being swept over the surface of the work. This scraper should be tapered from the middle towards the point and parallel from the middle to the heel, thus giving curved and straight cutting edges, which may be used for taking narrow or wide cuts, as the work may require.

FIG 103.

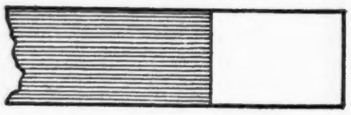


FIG. 104.

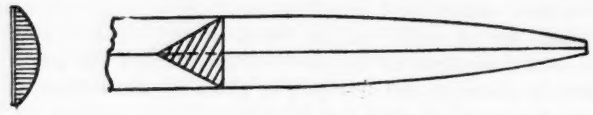


FIG. 105.

Work that is to have surfaces accurately fitted by scraping should be carefully planed. The finishing cuts should be very light ones, taken with a moderately fine feed, the work being clamped as lightly as possible, to prevent its springing when taken from the planer table. The surfaces should be filed only enough to bring them to approximate planes and to remove traces of tool marks.

The usual method of producing a plane surface is by comparing it with a standard plane. Such a standard is called a surface plate and bears the same relation to the testing of plane surfaces that the cylindrical gages do to the testing of circular surfaces. In Fig. 106 is shown a pair of Brown & Sharpe standard surface plates.

After bringing the surface of the work to an approximate plane by planing and filing, and too much care cannot be exercised in these operations, the work is ready for the scraper. A thin coating of red marking is rubbed over the face of the surface plate. The material used for this marking is usually Venetian red mixed in oil. Red lead answers fairly well, but separates too easily from the oil and does not spread as evenly and thin as the former. The marking can be best applied to the surface with the fingers or the palm of the hand, as the hand detects any dust or grit and spreads the marking thinner and more uniformly over the surface than can be done with a piece of rag or a brush. The work surface is now rubbed over the surface plate, the high points on the work being shown by the marking rubbed from the true surface of the plate. These high points, if small and few in number, may be reduced with a fine file until the work, when moved over the plate, will show fairly

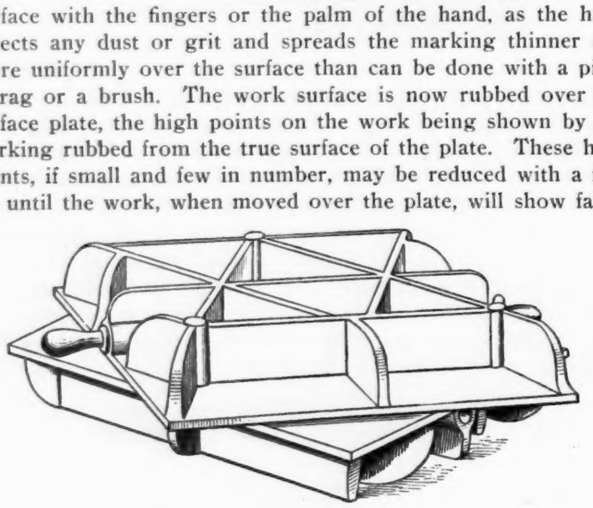


FIG. 106.

good contact. The file should be used up to that point in the operation at which more time would be required to make the file cut on the proper spots and sufficiently light to prevent pitting than would be required to remove the metal with a scraper. The workman's judgment must determine this point, and much time and hard work will be saved if his judgment is good. A file having considerable belly should be used for this purpose.

The thickness of the coating of marking will depend upon the condition of the surface, the nearer it approaches a plane the thinner the marking must be. If too thick false bearings will show, which lead to confusion and errors in the scraping. For the finest work it must be rubbed down so thin as to be scarcely visible, and the dark brown spots left on the work will then show true bearing points. The harder the surfaces are pressed together the plainer will the marks appear, the higher ones looking the brightest.

When the work is heavy and awkward to handle the surface plate may be rubbed over it. After each course with the scraper the surface must be remarked. For the first few courses the strokes of the scraper may be moderately long, never exceeding three-fourths of an inch, but as the surface becomes truer and the bearing points close together, the strokes must be made shorter, being careful not to overreach the marked points. Each course must be made at a considerable angle with the preceding one, thus preventing the waved surface that results from numerous cuts across the work in one direction.

The scraper should be held as shown in Fig. 107, and pressed firmly to the work. The pressure required depends upon the hardness of the metal being scraped and the condition of the cutting edge. This must be carefully considered in accurate finishing, since as the tool dulls the pressure must be increased in order to give cuts of equal depth.

The degree of accuracy required must in every case determine how far the process should be carried. For a strictly first class job there should be contact over practically the entire surface, as shown by the extremely thin coat of marking. Such surfaces are comparatively expensive to produce.

When two plane surfaces are to move over each other, as so frequently occurs with machine parts, both may be trued to the surface plate, but usually the nature of the work will prevent the use of the plate on both, in which case one surface may be trued to the plate and the other fitted to this surface.

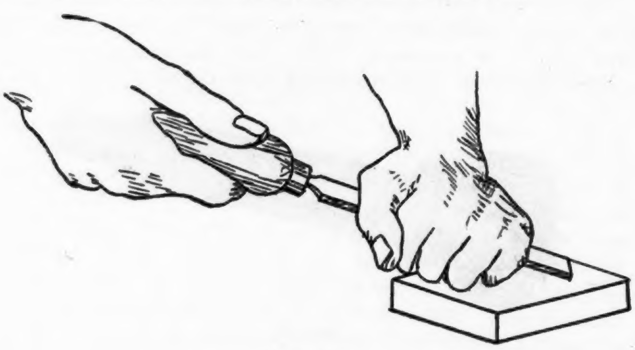


FIG. 107.

When two plane surfaces that are not intended to move over each other are to be fitted together one should, if possible, be trued to the plate and the other fitted to it, but frequently in such cases it is not possible to apply the plate to either surface, and usually motion of the one surface over the other cannot be had. In such cases the process known as "bedding" must be used. When possible one surface will be trued to the plate, then covered with marking and placed in position over the surface it is to be fitted to, when a sharp blow with a mallet or soft hammer will cause it to mark the high spots on the latter surface. These spots can be scraped away and the process continued until the marks appear uniform over the entire surface. If neither of the parts can be applied to the plate, one must be machined and finished as true as possible and the other fitted to it. This will, of course, not give true plane surfaces, but since in nearly all cases where motion between the surfaces is not expected a uniform bearing is all that is necessary, this method is satisfactory. As before, the coating or marking will depend upon the condition of the surfaces. If too heavy at any time the firer of the blow will spread it into the low spots, thus giving false bearings. Pedestal bearings, connecting rod brasses and similar parts must usually be fitted in this manner.

Cylindrical surfaces, as shaft, spindle and pin bearings, also depend largely upon the scraper for bringing them to their true bearing surfaces.

In such cases the bearing is usually fitted to its spindle, the latter taking the place of the surface plate. It follows that if

the spindle is not round a perfect bearing will not result. Since in machine construction the cylindrical truth of all spindles must be sufficiently exact to satisfactorily perform the operations for which the machine was intended, the bearings may be so fitted. In high grade machine construction the spindles are ground and lapped cylindrically true, thus enabling the skilled workman to scrape the bearings for these spindles as accurately as he could produce a plane surface from a standard surface plate.

In the absence of a standard plane a true plane surface may be originated in the following manner: Take three plates, the surfaces of which have been brought to approximate planes. Determine by means of the straight-edge which of these plates is the nearest true. Call this plate A and the others B and C. Assume A as a temporary standard, and fit B and C to it. The surface outline of A is shown to an exaggerated scale in Fig. 108. In Fig. 109 are shown B and C placed together after having been fitted to A. If now B and C are fitted to each other, being careful to correct equally the error on each, it is evident that either B or C will be nearer a true plane than A. Now select one of these plates, say B, as a temporary standard and fit A and C to it. Then fit A and C to each other as before. Next select C as the standard and repeat the process. Each repetition of this operation will bring the surface nearer to true planes, and when they finally interchange, showing perfect contact between any two the work has been completed.

Surface plates are designed to resist as much as possible the deflection due to their own weights. In large plates this is an extremely important point in their construction. For the final finishing the plates could be tested while standing on their edges, but if trued in this position they would sag in the centre

FIG. 108.

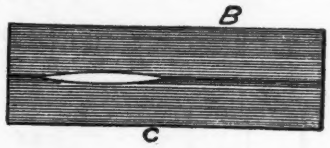


FIG. 109.

when turned down. It would, therefore, be necessary to apply them to the work in the same position in which they were finished, which would be extremely awkward.

A surface plate should always rest upon three points of support and should be kept in a substantial wooden box, from which the cover can be easily removed, exposing the surface when in use and protecting it when not in use from accidents due to articles falling on and marring the face, as well as from the dust. When not using the surface should be kept oiled to prevent rusting, which would impair its accuracy.

The surface plate should be kept in a temperature as uniform as possible and not varying far from that at which the plate was finished, as the expansion due to changes of temperature is very apt not to be uniform and the truth of the plate thereby affected.

In using the plate, wipe any dust or grit from the face before applying the work, otherwise there is danger of scratching the surface. A careful workman tests small work on the edges of the plate rather than in the centre, and thus prevents dishing the plate through wear.

* * *

Countersink the edges of all bolt holes and round off any other ragged corners that do not look well. It does not take long to do this, but it makes the difference between work that is finished and work that is not. Several fine engine lathes were recently received by a shop which were nearly perfect so far as alignment and fits were concerned. Every corner, however, was left sharp, and all the lathes had to be gone over with a file before they could be operated with any degree of comfort.

* * *

How many times have you found finished surfaces around bolt holes marred by the sharp corners of bolt heads or nuts that have been case hardened? It is seldom that there is enough clearance left under the corners, and when the case hardening takes place the bolt head or nut will spring enough to cut into the surface it is being set up against. Unless this very annoying but very practical point has been looked out for in manufacturing these small parts, the only way is to grind the corners before use.

A TOOL CHEST, ETC.

FRED E. ROGERS.

One of the ambitions of the apprentice is generally to possess a box in which to keep his shop possessions. It may vary from the modest pine affair knocked together at some noon hour with a hasp and padlock fastening to the black walnut cabinet fit for a parlor and costing considerable of his hard earned cash.

One of these creations is generally supposed to reflect great credit on the possessor, and to advance him materially on the road of promotion. Whether a mistaken idea or not, it is desirable to avoid extremes and to have a chest which is handy and in keeping with the ordinary shop surroundings.

Fig. 1 shows what might be called a chest of drawers, and is one that is easily constructed and not very expensive. The dimensions given are for one that is about right for the majority of cases. The drawers are separated by two sheets of boiler steel, 1-16" or 3-32" thick. These are set into the sides and back by sawing slots ¼" deep in the wood. The bolt of the lock in the lower drawer passes through the separating plate into the

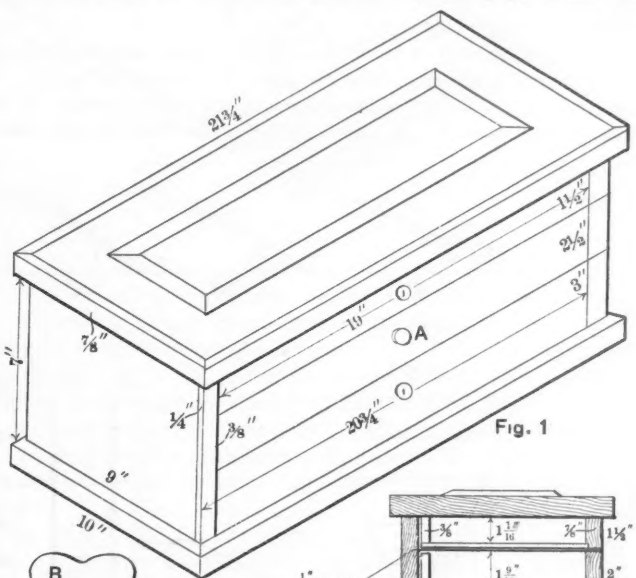


Fig. 1

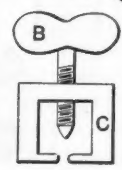


Fig. 3

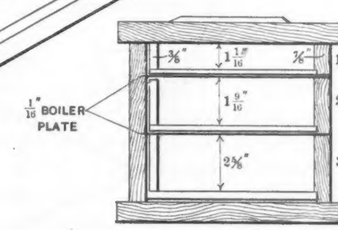


Fig. 2

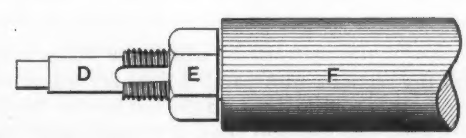


Fig. 4

second drawer and thus fastens the two, and the upper one is locked directly into the cover. By this arrangement two locks fasten three drawers and the tools commonly used can be kept in the lower two, and the more expensive and rarely used ones in the upper one, which can always be kept locked.

Fig. 3 is a tool for removing the hands of steam or other gages. These sometimes corrode in places and require considerable force to remove them, thus endangering the delicate mechanism within. By slipping this device over the hand with the screw in line with the end of the arbor, a turn or two easily shoves off the hand.

Fig. 4 is a scheme for tapping a hole in the end of a shaft or similar piece parallel with its axis, or, as commonly expressed, "square."

F is supposed to be of such a diameter that a square cannot be used on the end to test the tap, so a faced nut E is run on the tap D and used to test the accuracy of the work. The writer first saw it used by a machinist in tapping out the end of a long valve rod, and though simple it may be useful to others.

* * *

It is said that the Belgian Government has offered a prize of nearly $10,000 for the invention of a match containing no phosphorus. The competition is open to all nations, and there are a number of conditions that must be complied with.

SCREW PROPELLER PATTERN.

METHOD OF MAKING A PATTERN FOR A SCREW PROPELLER WITH SEPARABLE BLADES.

I. McKim Chase.

The first illustration shows a screw propeller different in shape from the one described in the March number of Machinery. The views are otherwise as well as more conveniently arranged, and they are numbered to follow consecutively those that have already appeared. The screw is a true one, that is, its pitch is uniform both radially and axially.

It will be observed that a blade developed on a plane is oval in shape, as shown in 134. The outer oval figure is intended to represent the shape of the blade as thus developed. Instead of the screw being cast entire the blades are made separately and provided with a flange for bolting them to the hub or boss. This is the type usually adopted for the propellers of ocean-going steamers. It possesses a distinct advantage when in need of repairs, for a broken blade can be renewed without removal of the screw from the vessel. However, a propeller of this kind is more expensive in first cost.

When a pattern is to be made of such a screw and the drawing, as is generally the case, does not show the developed sections of the blade, these should be developed by the pattern maker to full size, as explained in the March number. He should then determine the thickness of the material of which the blade is to be built and line off the sections with parallel lines according to the thickness adopted. It is well to dress up the lumber for the blade, rip it in pieces of sufficient size and allow it to stand until wanted for use. Next make two guide frames, one to conform to the angle of the pitch at the hub and the other to the angle of the pitch at the periphery. As the blade pattern is to be made with the face down ward, the angles of these guides will determine the proper helical form of the face of the blade.*

A substantial surface-board is required. It should be somewhat longer than the radius of the screw, and in width a little greater than the length of base of the largest guide. On this board draw a line through it lengthwise for a center line. With the radius of the screw and with one point of the trammel on the center line describe an arc across the board, the arc at the center line being about five inches from the end of the board. From the same center with a radius equal to that of the hub describe another arc across the board. Step off the length of the base of the triangle *e*, at 135, the periphery of which will be equal to the length of the base of the largest guide. Transfer this length to the arc corresponding on the surface board, making it equal on each side of the center line. From the extremities of the arc draw radii to the center, and the sector thus formed will be that with which a plan view of the blade will agree when its top and bottom edges are perpendicular to the axis of the screw.

* The method of making these guides was explained in the November, '97, number.

To determine the necessary length of the base of the guide when the outline of the blade is of special shape like that under consideration, it is necessary to lay down a projected view of the blade as viewed in line with the axis of the screw. Radial lines are then drawn tangent to the edges of the blade as at *f*, Fig. 134. The length of the arc at the periphery intersecting these lines is the fraction of the circumference. The blade extends through and is also the length of the base of the guide.

The two guides being prepared and having center lines drawn upon them, are to be secured to the surface board, their center lines and curvatures coinciding with the arcs drawn on the board; the guide on the periphery being on the inside of its arc. A pattern for the flange by which the blade is secured to the hub is required. It is to be secured to the surface-board by brackets, being properly situated in relation to the face of the blade as determined by the guides.

To facilitate laying off the pieces for the blade a templet like that illustrated at 136 will be found useful. It is made of thin stuff equal in length to the radius of the screw. The arcs of the different sections, as *a*, *b*, *c*, *d*, and also those of the hub and periphery are described upon it and enough of the templet cut away to admit of the arcs acting as guides while they are being marked on the blade pieces. The necessary preparations are now completed to allow the building of the blade to proceed.

It is not absolutely necessary with a blade pattern of this shape, where so much of it is cut away towards the periphery, to continue every piece to the outside guide. But the writer has found it good practice to do so. The small amount of material saved by stopping some of the pieces at the top and bottom short of the outside guide does not compensate for the extra care it involves to insure accuracy.

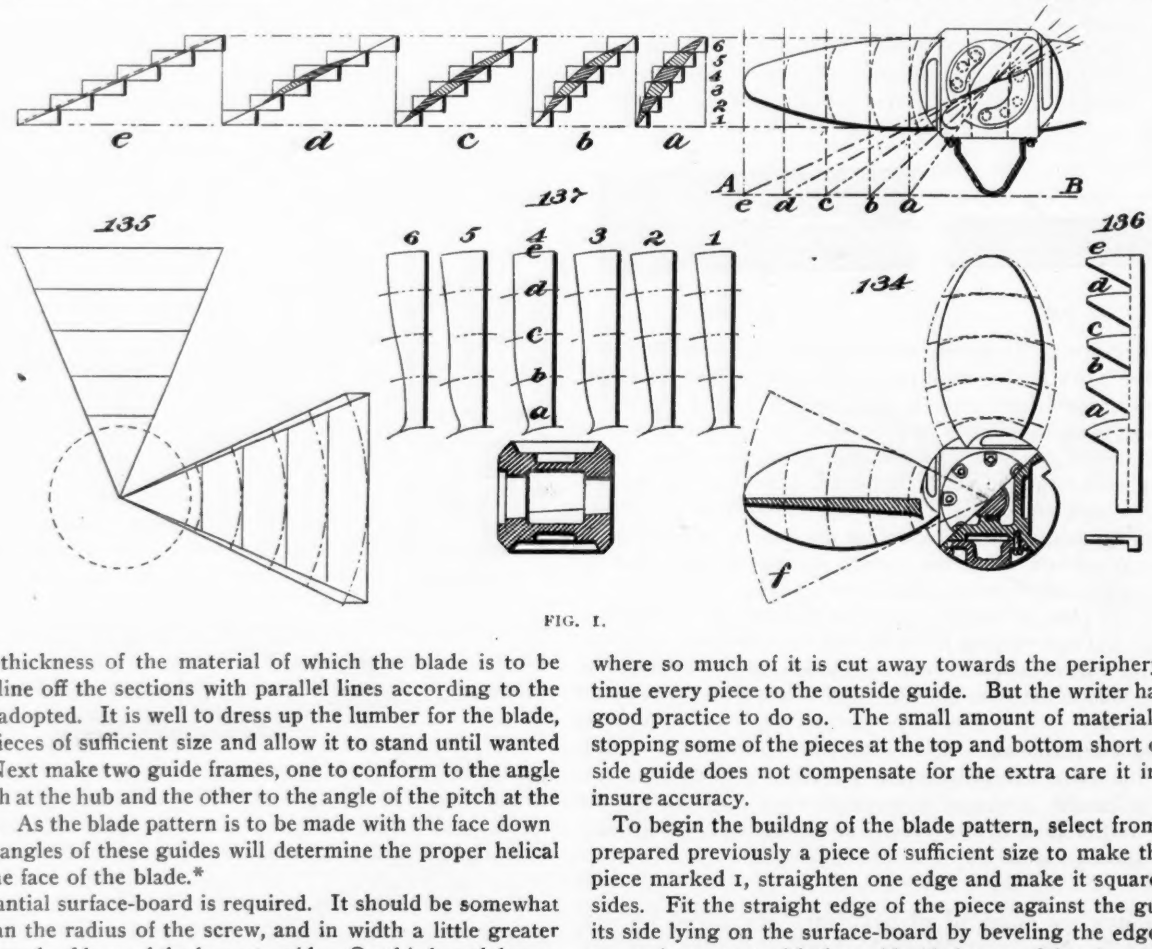


FIG. 1.

To begin the buildng of the blade pattern, select from the stuff prepared previously a piece of sufficient size to make the bottom piece marked 1, straighten one edge and make it square with the sides. Fit the straight edge of the piece against the guides with its side lying on the surface-board by beveling the edge where it comes in contact with the guides, being careful to have the bevels terminate exactly at the upper edge of the piece where it touches the guides. If it should occur in fitting a piece that the top of the bevel is carried in beyond the edge, the edge can be planed off until it coincides with the bevel. In this lies one of the advantages of fitting the pieces to the guides before reducing them to the shape in which they are built into the pattern.

Now mark the periphery of the blade on the piece which will be the outer edge of the outside guide; also mark the radius of the hub which is the inside edge of the inner guide. Lay the templet on the piece, the mark for hub and periphery coinciding with those of the templet, and mark the arcs of the different radii *a*, *b*, *c*, *d*, etc. Take the widths of the piece 1 at the different sections and lay them off on the piece 1. On the arcs corresponding to the section draw a line through these intersections with the aid of a batten, work off the edge to this line square with the sides, place

the piece on the surface board where it was fitted against the guides, securing it there against shifting, but in such a manner that it may be readily released when desired. Proceed in a similar manner with piece No. 2 which, when prepared, secure on the piece No. 1 by glue and with nails where they are not likely to come in contact with the tools in working the blade off. The remainder of the blade is similarly proceeded with until completed to the desired height. Where the parallel pieces come in contact with the flange they are fitting against it as well as against the guides. The shapes of the several pieces forming the blade are shown at 137.

After the pattern has been completed to the required height and before it is removed from the guides, it is to be roughly worked off on the back. Large, inside bevel gouges are useful for this purpose. The pattern is then to be turned over, with the face upward, utilizing the guides to hold it while the face is being worked off. When the face of the pattern is finished down to the lines formed by the joints of the pieces, the configuration of the blade is the next thing in order. This can be laid off directly on the face of the pattern, or a templet of the shape can be made of stiff paper and the pattern marked by it.

The shape of the blade being lined off on the face of the pattern, the surplus material outside of this boundary is to be removed. This accomplished, the face is given a coat or two of shellac and the pattern turned with its face downward and its back worked off down to the lines formed by the joints of the pieces of which the pattern is composed. The edges are next to be finished by working off the back to an easy curve where it trends toward the face.

When building up the blade it is well to avoid gluing the flange to the blade pieces. By arranging the flange to be removed while the blade is being worked off the latter is accomplished much easier.

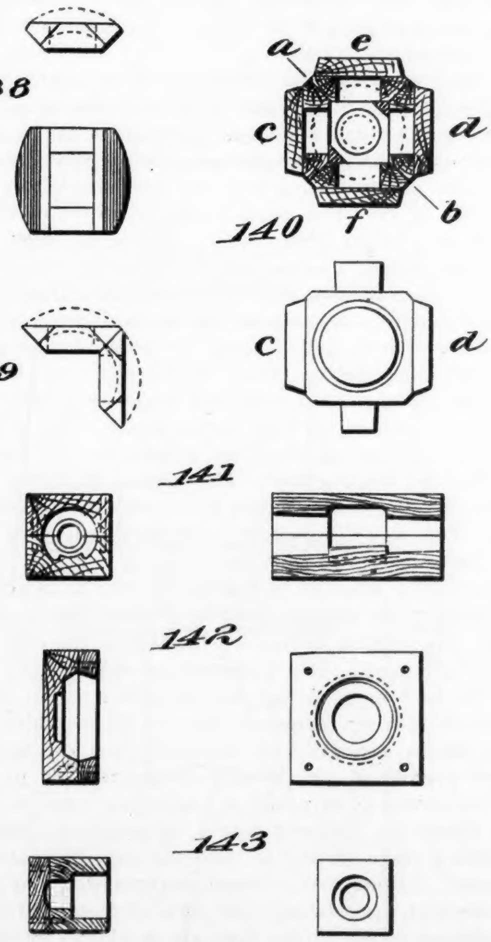


FIG. 2.

After the flange has been secured permanently to the blade, the fillets where they join are completed. The fillet on the back may be worked on the pieces which compose the blade, but the fillet on the face is best fitted separately.

The pattern, after being sand-papered and receiving several coats of shellac, is ready to be molded.

Glue alone should not be depended upon to hold the pattern together while being molded. Brads should also be freely used for the purpose.

Fig. 3 is an illustration of a partly built-up blade pattern. It differs in shape from those shown in the drawings. For the purpose of making prominent the manner in which the pieces of parallel thickness are fitted upon each other and to the guides they are shown somewhat disproportionate in thickness.

The pattern for the hub, or, as it is termed in England, the boss, is generally molded in loam when very large. When so molded the pattern is made a model of the casting; but is so constructed that it can be taken apart to permit of its withdrawal from the mold and the release of the interior parts or cores which have

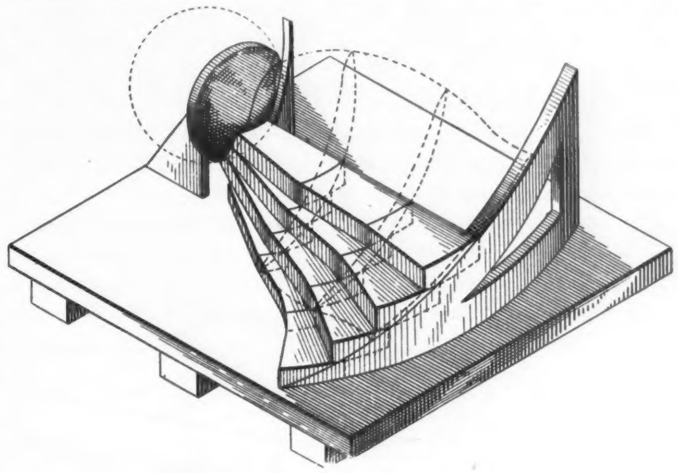


FIG. 3.

been made in the course of molding it.

When the hub is of moderate size it is best molded in dry sand. In this case it is first constructed in the form of a box which separates or parts diagonally across its ends. To form the sides four frames, as in 138, are made of stuff sufficient in thickness to allow of their being worked to the required spherical form. Each frame is made of four pieces. When these are joined together, they leave an opening in the middle of the frame which is covered by the core print for that core which forms the recess in the side.

The frames are fitted together with miter-joints (see 139). They are secured in pairs, each pair being glued and nailed together where they unite. The ends of the box, which are square in shape, are made in two pieces parting diagonally across the square. After these are secured to the sides corner blocks are glued and nailed inside the box to strengthen it at the corners where it is liable to be reduced to small thickness in being worked to the spherical form. Previous to securing the frames together their mitered ends are marked off by a templet having the radius of the hub and the material outside of this line is worked off square with the joint. When the four frames are put together the outline of the miter joints give the shape to which the pattern is to be worked in reducing it to the spherical form. The spherical form can either be obtained by turning in a lathe or by working the pattern off by hand. The writer has practiced both ways, but prefers the latter when the pattern is to be molded in dry sand and cores used to form the interior of the mold.

After the spherical form has been given to the pattern it is ready for the core prints. The prints, *a*, *b*, Fig. 140, for the tapered core which forms the shaft hole are secured permanently to the pattern; but *c*, *d*, *e*, *f*, on the sides, for the cores which form the recesses where the blades are secured to the hub, are made removable. They are held in place by draw-pins, which are withdrawn while the pattern is being molded, thus releasing the prints from the pattern and allowing the latter to be drawn from the mold first and the prints afterwards.

The recess cores are inserted from the inside of the mold previous to setting the main core, and are secured in the impressions made for them by the prints.

Fig. 141 shows a longitudinal and transverse section of the box for the core which forms the hole for the shaft through the hub. When but one casting is needed a half box can be made to answer; but when several castings are required it pays to make a whole box.

Fig. 142 shows a plan and a section of the box for the core which forms the recesses in the sides of the hub where the blades are bolted to it. Fig. 143 shows a plan and a section of the box for the core which forms the recess in the flange of the blade.

WHAT MECHANICS THINK.

A DEPARTMENT FOR THE EXPRESSION OF PRACTICAL IDEAS OF INTEREST, TECHNICAL OR OTHERWISE.

Write on one side of the paper only, and when sketches are necessary, send them. No matter how rough the sketches may be, we will see that they are properly reproduced.

A PULVERIZER.

One of the jobs of a man in a certain tool room is to pulverize prussiate of potash crystals to make a case-hardening preparation. To save time and incidentally save elbow grease, he devised the rig illustrated in the accompanying sketch to work the pestle in the mortar. A is a 6" wheel made from 1" board, with a piece of leather belting nailed more than half way around it.

The leather on A revolves in contact with the line shaft until the end of the leather reaches the shaft, when crank D, is a little below center, as in Fig. 1.

The pestle in connnection with cord E running over sheaves G and F, is now released.

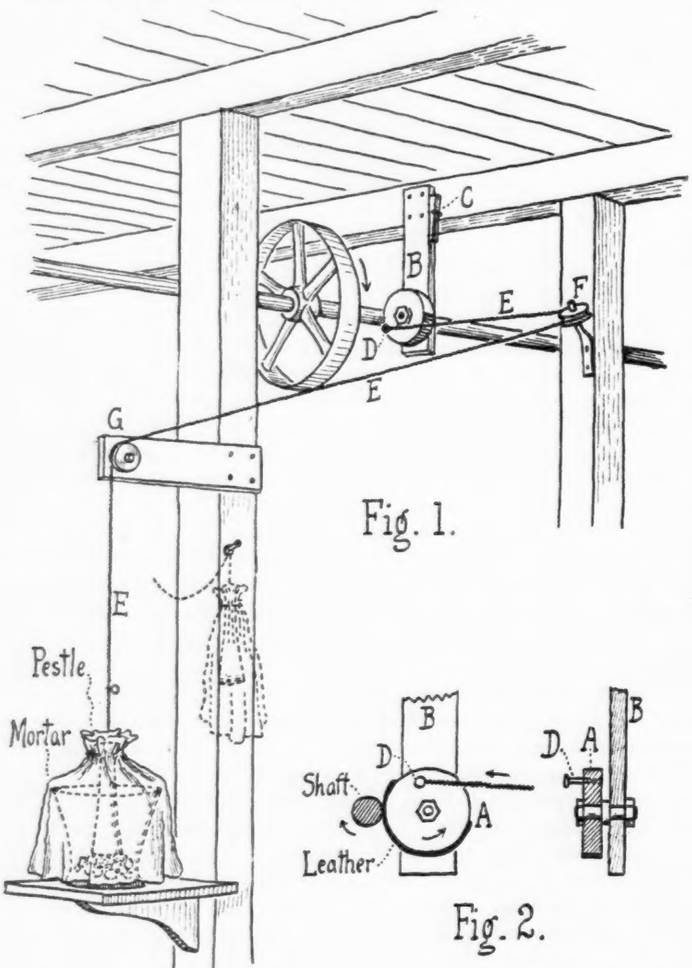


Fig. 1.

Fig. 2.

The momentum of A is sufficient to bring the first end of the leather in contact with the shaft when it acts, raises pestle and lets it fall continuously 50 or 60 times per minute.

When out of use the pestle is simply hung upon a nail in the post as shown, which leaves the wheel entirely out of contact with shaft. Cord E can be run to the most convenient place for the mortar. This works very smoothly, giving an easy lift and quick drop of about 4 inches to the pestle, which weighs 4½ lbs., and pulverizing thoroughly.

I did not sketch the tool maker sitting down watching this rig work, as he was busy with other work, thus saving money for the company. I hope I have made this clear.

Meriden, Conn. JAMES P. HAYES.

KEEP UP THE SPEEDS.

I have recently been through a number of iron working establishments in the vicinity of Boston and noted a great difference in them. Some are fitted up with special, modern tools for the manufacture of certain jobs, and it is a pleasure to go through such works and see the metal being cut off in a proper manner.

On the other hand, I have been through shops that are trying to get into this condition, and it is really heartrending to see the feeble attempts that they have made and the state of fear and awe in which they seem to stand regarding their machine tools, most of which are out of date, anyway, for fear they will break something.

As I think you have heard from me before, I have had charge of machine shop planers on cast iron at a cutting speed of 30 feet per minute, the return stroke being at the rate of 100 feet per minute, and kept this up day after day, taking a cut from 3-16 inch to ¼ inch deep, with ⅛ inch feed for a roughing cut, and a scraping cut with ¾-inch feed for finishing, and I have had no trouble in getting plenty of tool steel to stand this punishment.

In the old-fashioned shop of which I was just speaking, their planers were going from 8 to 10 feet per minute, and it was impossible (?) to have them go any faster. Their lathes and drill presses were run in the same manner. One man turning a 4-inch cast iron trunnion in an 18-inch lathe, had the back gears of his lathe in. I think that the cutting speed of that lathe must have been almost 4 feet (!) per minute, and if the foreman had insisted upon his taking out the back gears and speeding up his work to say 20 feet per minute, the man would have thrown up his job.

In another case a man was drilling ¾-inch holes with twist drills through cast iron boxes 4 inches thick. I figured roughly that the cutting speed of the lips of this drill must have been 2½ feet per minute.

I see no reason why the cutting speed of a lathe should not be as great as that of a planer. I can see the reason why the cutting speed of a drill should not be as great as the other two, because the drill is entirely surrounded by the metal being drilled and the heat due to cutting does not have the same chances of escape, but on the other hand a drill in cast iron should certainly be able to cut at the rate of 20 feet per minute without drawing the temper. As the circumference of a ¾-inch drill is 2.356 inches, or .196 foot, this drill should have run 102 revolutions per minute to get up to the 20-foot speed; but I think that nine machinists out of every ten would declare this to be a physical impossibility. That this is not true, however, is proved by the fact that I have had workmen under my charge drill and tap 320 ¾-inch holes in ten hours. This is thirty-two holes an hour, allowing, therefore, a little less than two minutes for drilling and tapping each. The tapping, of course, takes the greatest time, the drilling being accomplished in about one-half minute. This would represent 51 revolutions to each hole, and about 1-40 inch feed to each revolution of drill.

I am a strong advocate of pasting up over each drill or lathe a table showing the cutting speed for different diameters of work, with the belt position defined on the step of the cone with and without back gears. The workmen can see at a glance where to put the back gears in and out and where to put the belt on the cone for different diameters of work. These tables should be figured out for wrought iron, cast iron, steel and brass, and in ordinary conditions can be very closely adhered to. A thorough experience is necessary to determine what these proper speeds should be. It is, of course, impossible to figure out in the abstract these speeds, as they will not universally suit all conditions. I think that it should be figured out by each shop for themselves, considering their class of work and the quality and hardness of the metals they are working, which must be determined in each shop by experiment.

In the present days of economical methods of construction and economy of manufacturing operations, I think that "Machinery" can do no better work than to advocate such matters and keep them prominently before its readers.

Boston, Mass. WM. O. WEBBER.

A clear explanation of the Holly lathe problem, with sketch, has been received from Mr. C. Albert Wettengel, St. Louis; but as a solution was given in the last issue it will not be necessary to publish this one.

HE WAS A YANKEE.

Some years ago a fellow of my acquaintance rigged up a small steamboat, using an old sailboat for a hull and a discarded boiler with numerous patches and two or three flues plugged. For an engine he had an old rig that had seen its best days. It was of the spider pattern, with 3 x 5 cylinder. He did quite a business taking fishing parties up the lake. One day, when on one of these excursions several miles from town, he got careless, let the pump run too long, and as a consequence knocked out the top cylinder-head, breaking it in three pieces and breaking the flange on the top end of the cylinder like sketch No. 1.

Things looked unpromising for getting back to town that night, but the fellow had no idea of staying in the middle of the lake, so he set to work to repair the engine with the limited facilities he had on board, a hand saw and set of pipe tools, a monkey wrench, hammer and several short pieces of pipe and numerous pipe fittings. He first took an old tin pail used for bailing, and with a jack-knife and hammer cut the bottom out. This he placed on top of the cylinder, after removing the three broken studs remaining. He then sawed two pieces out of one of the seats, which were ash, 1 inch thick, two inches longer than the outside diameter of the flange on the cylinder. He also sawed another length off the seat and split off two pieces about 2 inches wide. He took the pipe-cutter and cut off the poker so he could use it for a burning iron. With this he made

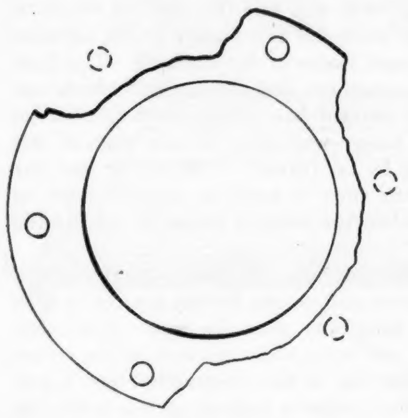


FIG 1.

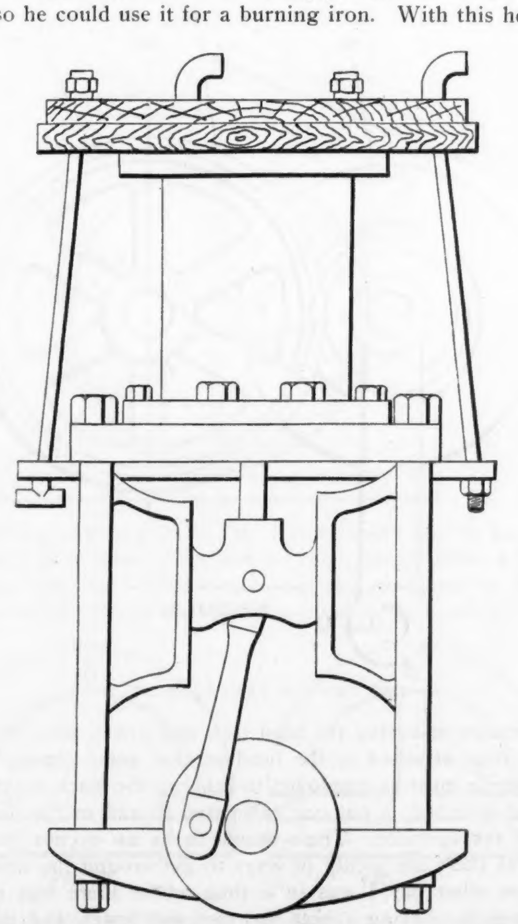


FIG 2.

holes in the boards to correspond with those in the cylinder flange. Taking a ¼-inch pipe he topped out the three holes in the remaining flange, cut some short pieces of pipe for studs and long pieces of pipe which he used as bolts, with bushings and elbows for nuts. Cutting a gasket out of the back of an old oil-skin coat he placed it on top of the cylinder. Next he put on the pail bottom, then the two pieces of board, placing them cross-ways the grain. He then took the two narrow pieces of board and put them under the cylinder baseplate. Using the four long pieces of pipe as bolts he screwed up the bushing and elbows on the top. Screwing the short pieces of pipe into the stud holes that he had re-topped, he used elbows and bushings for nuts. The repair completed looked something like sketch No. 2. Time, 55 minutes. He arrived in port that night in good season, and used the boat with this arrangement several times before it was convenient to get a new cylinder.

East St. Johnsbury, Vt. B. F. G.

[Although this tale may appear somewhat visionary, we are assured by the writer that it is a rehearsal of actual fact, and we have every reason to believe such is the case.—Editor.]

BUTTERFLY VALVES.

Many machinists have never been called on to make the kind of valve known as the butterfly valve. It is a nice job to make one that will not leak. These valves are not so much in use now as formerly, but sometimes we find a man who wants one.

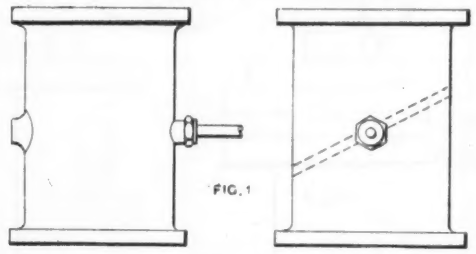


FIG. 1

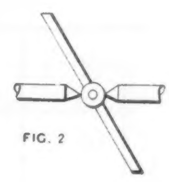


FIG. 2

The first thing is to bore the cylinder part true and round as possible. Then turn wing or fly part to drive neatly and tight enough to stay in place while the hole is bored through for the stem. It will be noticed that the wing is so placed in cylinder that there is no tendency to open during the boring operation.

After the hole is bored and the stuffing box finished drive the wing out and place it on the centers in the lathe and take a light finishing cut to size. I first take a light cut as far as the center of the hole, run carriage back without moving the tool, reverse the piece on the centers, and finish the other half. Clearance must be filled so that the wing can be turned in the cylinder. If sufficient care is taken the valve will not leak. I have made many valves of this kind to use in saw-mills for sawyer's valves; also larger ones to use as cut-off valves in exhaust pipes.

A.

PROBLEM FOR CARD SHARPS.

Editors: Enclosed I send sketch of a card taken from an engine designed by a famous engineer. The exhaust has considerable of a blow, and with quite a whistle to it. The steam chests are on the side and at either end of the cylinder, the steam pipe coming down between the two chests with a T connection. The engine has a riding cut-off valve, adjustable to suit. For a score of years crack engineers have tried to get a good card, but with the one result, a card like sketch. I enclose all the data obtainable:

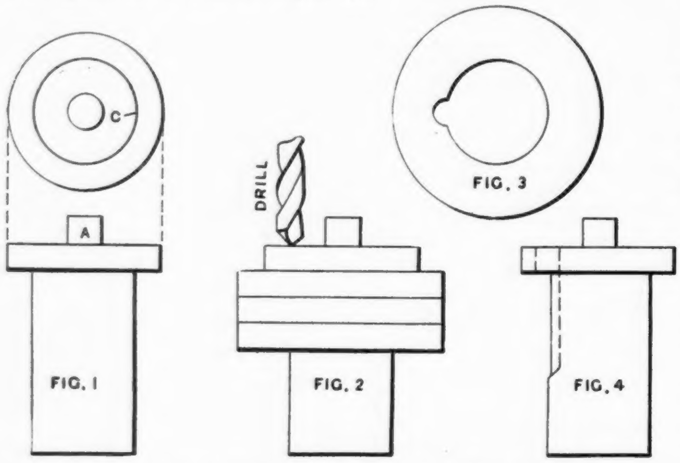
Steam pressure at engine, 60 pounds.
Distance from boiler, 150 feet.
Cut-off, 28 inches.
Revolutions, 50 per minute.
Cylinder, 15 x 48 inches.
Steam pipe, 3½ inches diameter.
Engine been running about twenty-five years.
The distance of engine from boiler is about 75 feet direct

but with the turns and elbows it doubles the distance. The engineer is an intelligent man and seems to know what he is talking about. I have my own views on the subject, but I want to know "what mechanics think." It is claimed that there was a mistake in designing the valves. W. De Sanno.

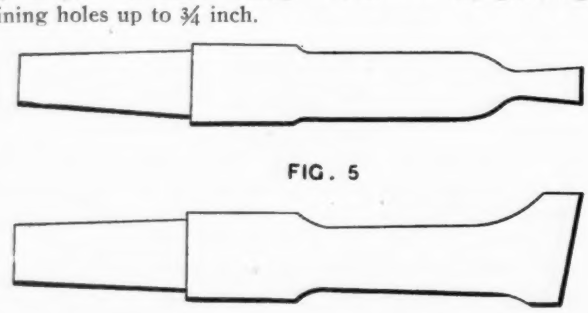
Tulare, Cal.

A SPLINING DEVICE.

In many shops where a spliner is not available, this kink may come in handy for splining small steel collars or sockets. To do the work, make a plug as shown in Fig. 1. This plug must be a drive fit in the collars or sockets to be splined. While the plug is on the centers in the lathe, the circle, C, is marked on the top face, of the same diameter as the drive part of the plug underneath the shoulder. This circle is to serve as a guide for drilling holes, as will be explained.



In Fig. 2 is shown the plug with three collars that are to be splined driven up against the shoulder of the plug. Now with a drill nearly equal in size to the width of the required spline, start on the circle, C, and drill down through the plug and the collars. This will take half the metal from the plug and half from the collars, so that, when the operation is completed and the collars have been taken off the plug, the collars will look like Fig. 3 and the plug like Fig. 4. This amount of metal having been removed from the collars, it is very easy to finish them with a splining tool in the drill press. The tool to be used looks something like Fig. 5, with a taper shank to fit the taper socket of the drill press spindle. Hold the collar in a slide rest and put a clamp on the drill press spindle to keep it from turning, and you will have a cheap as well as a very good rig for splining holes up to ¾ inch.



One or more, up to any reasonable number, of collars can be put on the plug and drilled at once; or the plug can be slipped into a socket that has to be splined. The small tip, A, in Fig. 1, is for removing the plug when it is used in holes where it does not bottom. Of course a new hole has to be drilled in the plug every time a new lot of collars is put on, but the same plug can the used for all collars of the same size until there is no more room for drilling. Horace E. Wood.

Hyde Park, Mass.

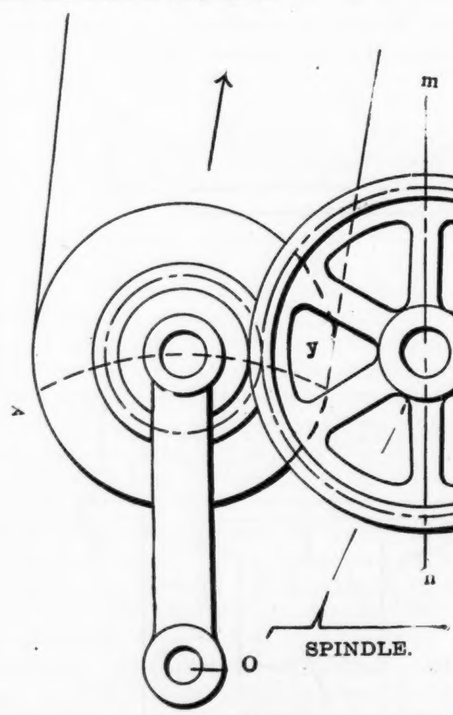
THAT MILLING MACHINE BLUNDER.

I was much interested in A. L. G.'s mention of the blunder in the design of a milling machine in his "Among the Shops" notes in the July issue. I have had a little experience with similar machines, and that experience has not been of the pleasantest character. I think, however, that in calling attention to the fact that the carriage has two horizontal swivels, for one of which he, nor any one else that he had seen, had been able to ascribe a reason, A. L. G. is not pointing out the worst feature of such machines, by any means. The design is probably an old one, and I doubt very much whether this, or similar machines, are now built—certainly not by progressive builders—and the contrast between this and later models represents the march of improvement.

The great disadvantage of this style of machine seems to me to be that the whole thing is overhung. There is no possibility of taking a good, solid cut without there being a give or a chatter somewhere. In fact, this seems to be the main feature of the design, and one is led to the conclusion that the designer must have had this idea in mind throughout and must have taken particular pains to have reached that end at every step.

Start with the head and see how nicely he has accomplished this object. The cutter arbor is supported at the middle, with the cutter at one end and the cone pulley and belt at the other—a first-rate see-saw arrangement and one that will be likely to improve with wear. Then there are two sliding joints between the cutter arbor and the main frame of the machine. The head moves up and down on an upright, and the upright moves out and in on the base. This upright has a very short bearing on the base and the cutter hangs way over to one side of this bearing so that there can be no reason in the world why the cutter should not jump any time it wants to and as much as it wants to, particularly when the head is raised up to the full extent of its travel.

Continuing to the table, here are two more overhangings. The table, slide, index device and all the fittings are on an arm of slim proportions and hang way over the base. The table, also, is supported by this arm for a short distance at the center only, and when it is fed clear out on the longitudinal feed it gets a double-extra overhang that makes a regular spring board out of it. If you want to see some fun, run the head clear up and the table clear out and try to mill something that has to pass inspection. A pair of micrometers won't be needed to tell how much the work is out, either.



THE NEWTON ARRANGEMENT.

The feature of having the head raise and lower, with the main driving cone attached to the head so that some clumsy tightening device must be employed to take up the slack every time the head is raised, is not one calculated to add to the peace of mind of the operator. There seems to be no excuse for such a plan, as there are plenty of ways to get around the difficulty. Only the other day I was in a shop where there was an old Newton miller having a head to raise and lower, and the belt arrangement impressed me as being very well adapted to the situation. It is shown in the accompanying sketch, and while it may be well known to many, I am sure it will be new to a few. Instead of belting directly to the spindle, there is a large gear on the spindle, as shown, and this is driven by a pinion

back of it on the same shaft with the driving cone. The cone shaft is carried by swinging arms, pivoted at point O, and as the cutter spindle is moved up or down on the line, m n, the cone, cone shaft and gear are free to swing along the arc, x y. The pinion can always be kept in contact with the gear, and there is no slack to take up in the belt. There can either be a spring or a weight fixed to insure the pinion keeping in mesh with the gear, or the same result could be accomplished by setting the countershaft a little to one side, so that the pull would come in the direction of the arrow in the sketch. I did not notice what method was used in this particular machine.

OLIN SNOW.

Philadelphia, Pa.

A SAW KERF FOR PATTERN MAKERS.

A somewhat novel feature in band sawing was performed, which may not be entirely new, but which seems to be difficult to do at first sight. A pattern was wanted for casting a pipe

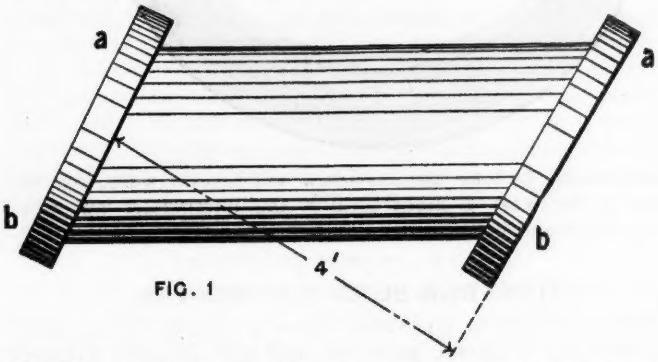


FIG. 1

like Fig. 1. A piece of pipe pattern of the required diameter was used to form the body, and the flanges were sawn out on the band saw to the correct bevel, so that when they were straddled over the body of the pipe they came at the right angle. The method of sawing the flanges was as follows:

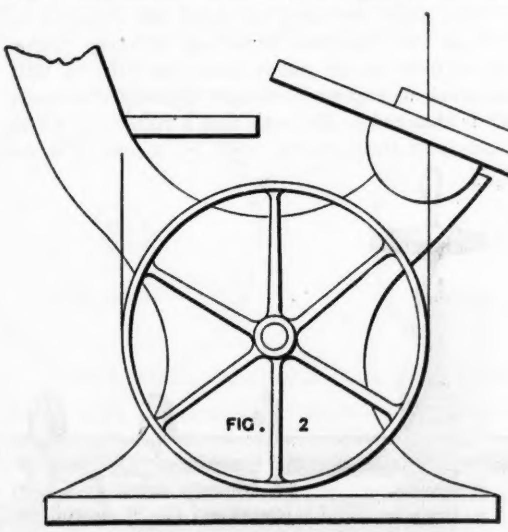


FIG. 2

As can be observed from Fig. 1 the flanges had to have the bevel sawn in opposite directions on the opposite sides, a and b. The band saw had a table tilting one way, as shown in Fig. 2. For one-half the flange the table was tilted to the required angle,

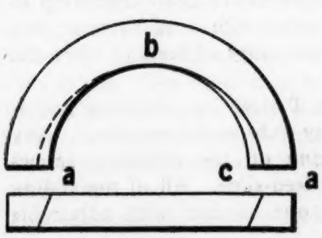


FIG. 3

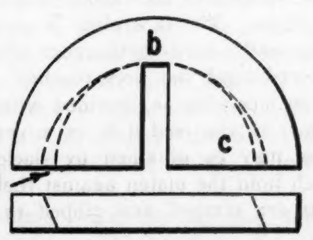


FIG. 4

as in Fig. 2, and beginning to saw at point a, Fig. 3, the table was gradually lowered to a level plane while the saw was running. At point b the table was brought to a horizontal position and the flange was then turned bottom side up, the table again tilted, and the sawing started at c, and terminating at b, the table being lowered the same as before. This completed the first half of the flange. To saw the other half, a space a trifle wider than the width of the saw was cut in the block, so as to start from b, Fig. 4. The table was level when making the start at this point, and was tilted during the sawing to give the proper angle when c was reached. For sawing from b to a the block was turned over on to the other side and the sawing was done the same as from b to c, starting at b.

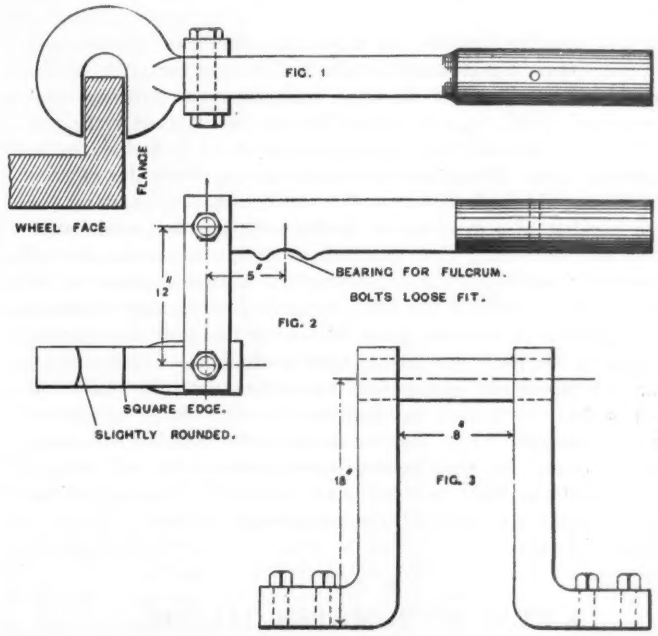
B. SAWYER.

Buffalo, N. Y.

PRY-OVER JACK.

A convenient pry-over jack for large fly wheels is shown in the accompanying sketches. It is intended for wheels having a flange around the sides of the face, but by modifying it a little the same design can be used for wheels having no flange, provided the rim is enough wider than the belt to let the clamp get hold. The necessary changes to accomplish this will suggest themselves to any engineer.

In Fig. 1 is a plan view showing the clamp or dog in position for gripping the flange. In Fig. 2, which is a side view, is shown how this clamp is connected by two links, or side pieces, to the main lever. The fulcrum upon which this lever rests appears in Fig. 3, and its construction will be clear from the sketch. The top bar can be made of 1½ inch cold-rolled steel.



The clamp should be made of 1¼ inch or 1½ inch machinery steel, and the two side pieces should be of sufficient width to take ⅞ inch or 1 inch bolts, these bolts being a loose fit. The lever should be made of machinery steel and have the outer end rounded to fit inside of 1½ inch gas pipe, this making a strong and light-weight handle. The length of the lever and pipe together will have to depend upon the available room, but it would be best to have them from 7 to 8 feet long, and the pipe should be pinned to the steel lever.

To make sure that the clamp will let go when the lever is raised or when the wheel starts, it would be well to make the slot in the clamp where it grips the flange about ⅛ inch wider than the thickness of the flange. The fulcrum is to be fastened permanently to the floor, and can be easily removed and laid one side out of the way when not in use. A pry-over jack made of the proportions indicated will answer for an engine of from 1,000 to 1,200 horse power.

FRED COLLINS

Providence, R. I.

KNURLED SCREW HEADS.—TEMPERING SMALL DRILLS.

The edge of a screw head or other circular piece can be very effectively knurled when no knurl is at hand by using a mill file. Roll the screw head back and forth under the pressure of the file held firmly in both hands. The lower edge of the screw head should be upon hard wood.

A very simple and effective method of hardening and temper-

ing small drills of about 1-16 inch diameter and under, is to heat the drill to a cherry red and immediately plunge it into a ball of beeswax.

GEO. H. WALTMAN.

Hokendauqua, Pa.

DIAMETRICAL PITCH AGAIN.

After reading articles that have appeared from time to time, on subjects that are as clear as day to an engineer, it seems as though something should be done to correct the false impressions that exist in the minds of shop foremen, etc., from the West and everywhere else.

To illustrate this point, the writer will refer to an article contributed to the July issue of "Machinery" under the imposing had of "Diametral Pitch." If any one is sufficiently interested in this small section of gearing to investigate what is meant by diametral pitch (and well might "Mr. Foreman" do so) the surprising simplicity of the matter will make a "self-made engineer" feel cheap.

If our august correspondent, "Mr. Foreman," will consult either "MacCord's Kinematics," or "Unwin's Machine Design," the idea that he has obtained of diametral pitch will receive the quota of "reform treatment" that it so sadly needs.

"The diametral pitch of a gear wheel may be defined as that fraction of the diameter that the circular pitch is of the circumference."

It has become common to drop the numerator of the fraction expressing the diametral pitch, and denote the pitch by the denominator only. For instance, take the very problem that "Foreman" cited. A gear wheel having twenty-four teeth and is 3" in diameter will have a diametral pitch of 3" divided by 24 equals 1/8 inch. Dropping the numerator we have the expression "8 pitch" which "Foreman" himself uses. Expressing the pitch number of a gear wheel in this way, has led to the mistaken idea of calling that number the diametral pitch. It will be evident, therefore, that expressing the diametral pitch by the denominator instead of the whole fraction is equivalent to using the reciprocal of the rule given above, or dividing the number of teeth by the pitch diameter, which is obviously wrong. So if your correspondent will please remember that the diametral pitch is that fraction of the diameter for each tooth, instead of the unreasonable ratio that he propounds (and is sustained, strange to say, by well known manufacturers) he will take a step toward the light of "why and wherefore," instead of remaining at the station of "It's-good-enough-for-me."

New York City. L. S. LEVY.

* * *

A NEW MICROMETER CALIPER.

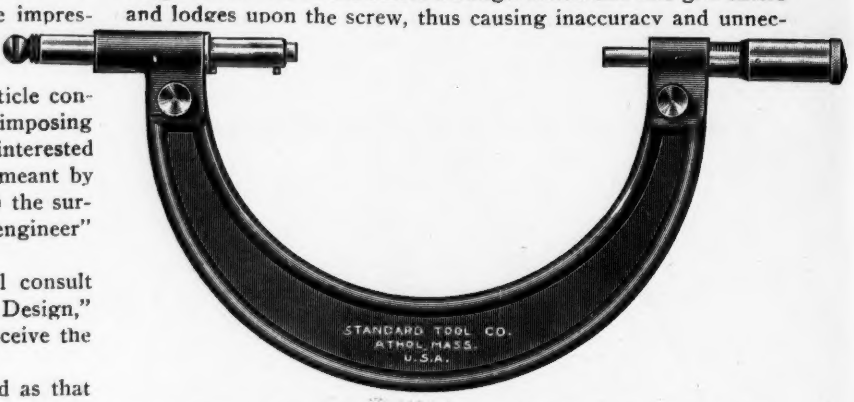
The Standard Tool Company, of Athol, Mass., are placing upon the market a micrometer caliper which is made in two sizes, one of which will caliper from 0 to 3 inches, and the other from 1 inch to 6 inches, thus making but two instruments necessary for all sizes up to 6 inches. The frame is of steel and of the usual horse-shoe shape, but with rims on the outer and inner edges to secure greater rigidity. The anvil end is fitted to take an adjustable anvil 3 or more inches in length, having hardened stops accurately set at intervals of 1 inch. The stops rest upon a hardened steel facing on the frame, and when the anvil is once set it is firmly locked in position by a knurled-head screw passing through the frame.

The measuring screw has a range of 1 inch, as in the ordinary 1-inch micrometer, and in order to change the range of the instrument from 1 inch to that desired, the thumbscrew in the frame is loosened and the anvil moved to the required stop, when a slight turn to the right will bring the stop firmly to its seat, where it can be again clamped. The measuring screw can also be clamped, making a fixed gauge of the instrument.

An important feature of the measuring end of the caliper is that the sleeve which surrounds the thread of the screw can be removed at any time for cleaning. Instead of being riveted to the screw, as in the usual construction, it is securely fastened to it by a clamping device, and mechanics who have had experience in trying to clean their micrometer screws by forcing them through the nut will appreciate this removable feature.

Another feature of this construction is that when the face of the anvil becomes worn, the tool can be quickly and accurately adjusted by twisting the sleeve upon the shank of the micrometer screw and setting the 0 lines so they coincide. This leaves the face of the anvil and screw parallel, and is believed to be an improvement over the old method of turning the anvil to take up the wear, from the fact that when the anvil is turned it is likely to throw the faces out of parallel.

The barrel is one solid piece, in the upper end of which is located the micrometer nut so arranged as to overcome the usual longitudinal slits in the barrel through which dirt and grit enters and lodges upon the screw, thus causing inaccuracy and unnec-

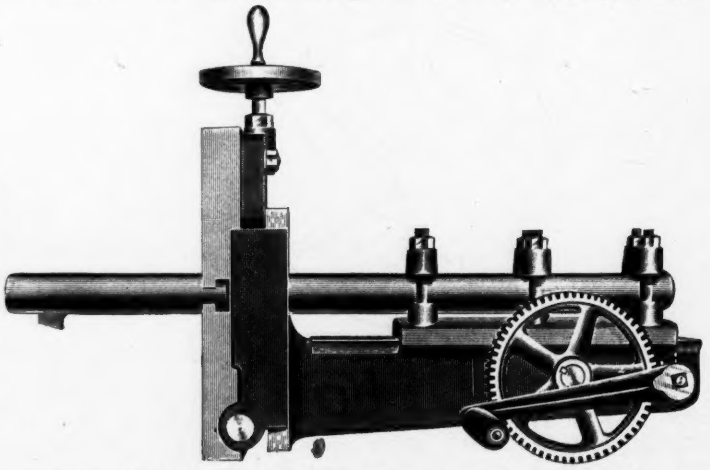


essary wear. In long use this screw will become worn, and to take up the wear a slight turn of the micrometer nut will compress it and make it as good as new.

* * *

PORTABLE BORE KEYSEATER.

The accompanying illustration shows a machine for keyseating the bores of pulleys, gears, etc., and will cut either straight or tape keyways up to 1¼" wide by 5/8" deep in any bore not longer than 12". It can be used on the bench-legs furnished with the machine for all wheels up to 24" in diameter, the work being chucked to the platen, which is provided with T slots for the purpose. When operating the wheels of larger diameter, it will usually be found more desirable to chuck the machine to the bore, thus saving the expensive handling of heavy pieces, the keyseating being done as the pieces leave the lathe or mill. The machine will work in any position, and consists essentially of a platen, which is chucked to the bore, and a cutter-bar which is moved back and forth through the work by means of a cut



rack attached to the saddle, and a set of cut gears operating in the frame. The face plate is provided with a micrometer feed gauge and a total depth gauge which stops all feeding when the required depth has been reached.

The face-plate is provided with T slots for chucking and is hinged to the feed-slide in a very substantial manner. Any taper may be obtained by slacking off the clamping screws which hold the platen against the feed-slide. All of the sliding parts are scraped and gibbed to one another with adjustable gibs.

Full sets of cutters are furnished with the machine when desired, but they can very easily be made by any tool maker. It is found that this machine will often relieve more expensive machines from the work of keyseating, and it is put out as a companion to the portable shaft keyseater made by the same firm, John T. Burr & Son, Kent Avenue and South Sixth Street, Brooklyn.

THE PRINCIPLES OF BELTING.

THE FACTORS THAT INFLUENCE BELT CALCULATIONS.—WRITTEN FOR YOUNG MECHANICS.

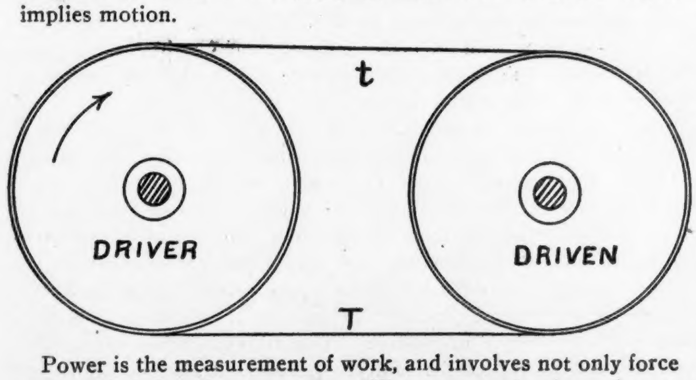
RETSEL.

Several years ago Prof. Gaetano Lanza read a paper before the American Society of Mechanical Engineers upon the subject of belting, in which he compared a number of the belting rules in use, by computing by the various rules the width of belt necessary for transmitting 30 horse-power, when traveling 1,500 feet per minute over 30-inch pulleys. Eleven rules were used and the results varied from 12 inches to 43 inches for the width of the belt. Evidently all of these cannot be right, though widely varying widths might answer the purpose under suitable conditions.

This may exaggerate a little the statement of the belt transmission problem, but it is certain that there are many rules for belting that do not agree at all. Some of these are based upon experiment, and some have no good reason for existing, though it is only fair to suppose that all of them represent the experience of somebody working under certain special conditions. In belt calculations, therefore, it is well to be able to select a rule intelligently, and to do this the general principles of belting must be understood. It is the purpose of this article to explain some of these principles and to point out just when we can rely upon these in proportioning belts, and when we must rely upon our judgment and our knowledge of the circumstances in hand.

The Pull that Does the Work.

In the first place, it will be well to remember what is meant by the words "Force," "Work," and "Power" in mechanics. These are words that are often met with in anything that is written about belt transmission, and the distinction between them is not always made. Force means a push or a pull, such as we exert with our hands or our arms. When a force acts through a distance, as when we push a table across the floor, work is done. One might push indefinitely against the side of the house without doing any work in the mechanical sense, but let motion occur, as in the case of the table, and work will be done. Work implies motion.



Power is the measurement of work, and involves not only force and motion, but time as well. When we speak of one horse-power, we mean an amount of work done in a minute equivalent to lifting 33,000 pounds (the force) one foot high (the distance) in one minute (the time).

In Fig. 1 are two pulleys, the driver and the driven, connected by belt. The belt will, of course, be put on with a certain tension, which we will represent by the letter t. Before the shafting is started, both the upper and lower parts of the belt will have the same tension. Both parts will pull with a force of "t" pounds, and tension "T," as shown in the cut, will be equal to tension "t." Now let the driver start in the direction indicated by the arrow. The tension in the lower side of the belt will increase, while the upper part will become slack and its tension will be less than formerly. This will go on until the difference in the tensions is sufficient to start the driven pulley, or until the belt slips.

The difference in the two tensions, T-t, is the pull that does the work in power transmission, and is called the mean effective pull. This is the effective force that acts in transmitting power from one pulley to another and one or two illustrations will show its importance in belt calculations. Suppose, for example, that a belt traveling with a velocity of 1,500 feet per minute transmits 10 horse-power. 10 H.P. = 10 ×33,000 = 330,000 foot-pounds of work done in one minute, and as the effective pull multiplied by the number of feet passed through in one minute must also equal the foot pounds of work done in that time, the effective pull will be equal to 330,000 ÷ 1,500 = 220 pounds.

Again, suppose the tensions on the tight and slack sides of a belt traveling 2,000 feet per minute to be respectively 500 and 200 pounds. Then the effective pull will be 500 — 200 = 300 pounds, and the foot-pounds of work done in one minute will be 300 × 2,000 = 600,000. The horse-power will therefore be 600,000 ÷ 33,000 = 18.88.

From what has been stated above, it will be clear that when power is being transmitted the pull or tension in the tight side of the belt is greater than the effective pull that does the work; also, that this tension is greater than the original tension given to the belt when it is put on to the pulleys, before they are started in motion. It should be mentioned, however, that the sum of the tensions, T + t, remains practically the same, whether the pulleys are at rest or in motion. Thus, if a belt is put on with a tension of 500 pounds, both T and t will equal 500 pounds and the sum of the tensions, T + t will be the pull upon the shaft. After the shafting starts, tension T will increase, but tension t will decrease a like amount, so that the total pull on the shaft will remain the same.

Factors that Determine the Width of Belt.

The main factor that determines the width of a belt is its strength and wearing qualities. A belt should be wide enough to bear for a reasonable length of time the greatest tension that will be given to it without undue straining or stretching. This is the tension, T, of the tight part of the belt. Information upon the strength or endurance of belting is not very plentiful, and the strength in any case depends upon the kind of fastening used. If a glued joint is used, the belt will be practically as strong at the joint as at any other part of the belt; but even here it would not be best to allow too high a working strength, since it might result in stretching to such an extent that the slack would have to be taken up sooner than should be necessary where it is so much trouble to make a new joint as is the case with a glued belt. It may be safely assumed that a single leather belt, one inch wide, will sustain 200 pounds through the lace holes before breaking and that one-third of this, or 67 pounds, is a fair working strength. In any case these figures will serve as a rough guide in arriving at a suitable working strength for a belt.

Besides the greatest tension, T, of the tight part of the belt, three other factors enter into the determination of the width. They are, the mean effective pull, the coefficient between the belt and pulleys and the size of the arc of contact of the belt on the smaller pulley. The effective pull has already been touched upon and can always be easily arrived at. The coefficient of friction depends upon the kind of belt and pulleys used, and the character of the surfaces in contact. The coefficient varies widely and according to no particular rule. For ordinary leather belting, running on cast iron pulleys with no more than 1 per cent. of slip, .30 is a fair value, while certain conditions might make it vary from .20 to .50. The arc of contact on the small pulley influences the results from the fact that when a belt runs from a large pulley to a small one, it encircles only a small part of the small pulley and slipping will occur there before it will on the large pulley.

These four factors, T, T-t, the friction and the arc of contact are closely related and their relations can be expressed by equations; but as these equations are not of a very simple nature, the following table has been prepared, which shows at a glance what these relations are:

TABLE SHOWING GREATEST TENSION (T) IN THE BELT FOR ONE POUND MEAN EFFECTIVE PULL.

Angle of Contact of Belt on Smaller Pulley. Degrees.	Co-efficient of Friction of Belt on Pulley.			
	.20	.30	.40	.50
60	5.35	3.71	2.92	2.45
100	3.37	2.45	1.99	1.72
140	2.65	1.92	1.61	1.42
180	2.15	1.64	1.40	1.26

The table shows the greatest tension, T, in the driving part of the belt for one pound mean effective pull. Values are given for various lengths of arc of contact on the small pulley, and for coefficients of friction ranging from .20 to .50.

A few examples will illustrate the use of the table.

1.—Find the width of single belt needed to transmit 15 horse-power, at a speed of 2,500 feet per minute, when the arc of con-

tact on the small pulley is 100 degrees. First compute the mean effective pull required to produce 15 horse-power at the given speed: 15 horse-power = 15 × 33,000 = 495,000 foot-pounds of work done in one minute; 495,000 ÷ 25,000 (the speed) = 198 pounds, mean effective pull. We will assume .30 as our coefficient of friction, and hence in the table under .30 and against the arc of contact, 100, we find the number 2.45. This gives us the greatest stress that will occur in the belt for every one of the 198 pounds of mean effective pull. As we are to proportion the belt according to its strength, we find as the total stress that the belt will have, 198 × 2.45 = 485 pounds. Assuming that the belt will stand a stress of 66 pounds per inch of width, the value mentioned above, the required width of belt is 485 ÷ 67 = 7.2 inches. A 7-inch belt would be used.

2.—Again, find the horse-power that a 1-inch single belt will transmit when traveling 800 feet per minute and with an arc of contact of 180 degrees, or half the pulley. The process is evidently just the reverse of that in the last example. The allowable tension for the belt is 67 pounds, and this divided by 1.64 taken from the table gives 40.8 as the mean effective pull; 40.8 × 800 = 32,640 foot-pounds in one minute, or nearly one horse-power; one horse-power being 33,000 foot-pounds.

General Rules for Belting.

This gives us at once a general rule for belting that is used by many engineers and is very convenient to remember. It is that a single leather belt one inch wide, traveling 800 feet a minute, will transmit one horse-power. This rule assumes that the pulleys are of about the same size. In case there is considerable difference in their diameters, or for any other reason the rule is not thought sufficient, the speed can be taken at 900 feet per minute, or even 1,000 feet per minute.

Upon this basis of 800 feet per minute, the following formulas may be used, it being remembered that it assumes pulleys of about the same diameter.

Let H = horse-power transmitted,
W = width of belt in inches,
S = speed in feet per minute.

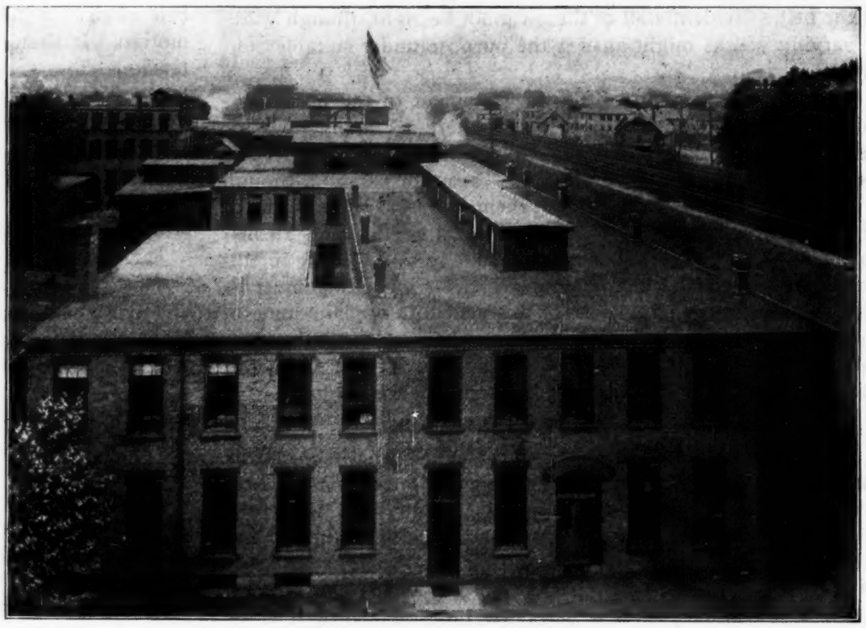
Then $H = \frac{WS}{800}$, $W = \frac{800\,H}{S}$ and $S = \frac{800\,H}{W}$.

At the beginning of this article it was stated that in using a rule for belting we could depend somewhat upon general principles, but that it is largely a question of judgment. We have seen that there are certain relations existing between the tension in the belt, the effective pull and the friction. A knowledge of these relations gives a knowledge of general principles. Upon our judgment in any case must depend what values we assign for the coefficient of friction and for the strength or durability of the belt. There is one other condition that has not been mentioned, the tension with which the belts are put on the pulleys. The time may come when people will put on their belts with a known tension, but it is not yet at hand, and if a belt is proportioned correctly all that is necessary is to put it on to the pulleys with sufficient tension so that it will not slip to too great an extent. As it now stands, this is a matter of practice and not of theory.

In proportioning double belts the same principles should hold as in calculations for single belts. Under ordinary conditions, if a double belt is twice as strong as a single one, it ought to be capable of transmitting twice as much power, or even more, if, as is commonly the case, the joint is glued. It is doubtful, however, whether a double belt would be put on with sufficient tension in ordinary practice to prevent slipping, if proportioned according to the assumption that it is twice as strong as a single belt. It is a question whether it would have twice the endurance and whether it would hold its tension for a great length of time if put on in this way. It would be safer to assume a strength of from 1½ to 1¾ the strength of a single belt and thus increase its durability.

THE CHIMNEY WAS NOT NEEDED.

The chimney has so long stood as one of the evidences of the existence of a manufacturing plant that it surprises one to look out upon an area of buildings evidently devoted to manufacturing purposes, and not see the tall shaft of brick pointing heavenward and usually capped with a cloud of smoke. The chimneyless establishment of the B. F. Sturtevant Co., at Jamaica Plain, Mass., however, serves to awaken such surprise, for it is equipped with a mechanical draft plant, and the otherwise useless chimney has just been torn down for the sake of the bricks



A FACTORY WITHOUT A CHIMNEY.

it contained. It will be remembered that this plant was quite fully described in these pages by Mr. Walter B. Snow, in his series on mechanical draft, and the Sturtevant Co. have sent us the accompanying photograph, showing the factory as it appears without the chimney.

When the tracks of the N. Y., N. H. & H. R. R. were recently elevated, it became necessary to make extensive changes in the arrangement of these works, which are adjacent. Among other alterations, the location of the boiler plant was changed to a point so far distant from the chimney, which had produced the draft, that its further use was precluded. This condition proved, however, a most excellent opportunity for the introduction of mechanical draft in the success of which this company has been deeply interested. Accordingly a Sturtevant fan was installed on top of the boilers, the gases drawn through it and discharged through a short stack extending but a few feet above the top of the boiler house. The arrangement has proved to be simple, economical and convenient.

Although the present stack, which can be dimly seen in the photograph, does not extend above the level of the surrounding buildings, no inconvenience has resulted therefrom. In fact, smoke is scarcely ever visible, and then only for an instant, the positive and ample supply of air resulting from the operation of the fan, serving to promote perfect combustion. By means of a special automatic device, the speed of the fan is exactly regulated to the requirements of the fire and the steam pressure is maintained absolutely constant.

* * *

Liquid air, the latest fad in engineering, has been hunting around some time to find a use for itself. It has the ability to produce extremely low temperatures, but having got the low temperature, no one seemed to know what to do with it. A Russian, however, has already begun to experiment with it. He placed a dog in a room with the temperature lowered to 100 degrees below zero, and after ten hours the dog was taken out alive and with an enormous appetite (so it is said). Then the physician tried the test himself. After ten hours' confinement his system was intensely stimulated, and so much combustion had been required to keep warm that an intense appetite was created. Here is a chance for mechanics to make money. Set up a compressed air plant and sell people appetites.

HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

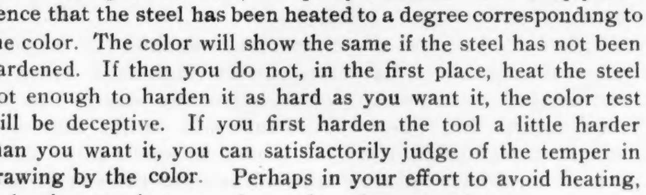
Give all details and name and address. The latter are for our own convenience and will not be published.

112. R. S. E. asks: How can I give small iron articles a thin coating of copper? I have frequently seen articles thus coated and should like to coat some myself. A. Thoroughly clean by immersing in sulphuric acid, diluted with water, and scour and wash. Then immerse in a fairly strong solution of sulphate of copper. When sufficiently coated remove and wash in clean water.

113. A. V. L. asks: What is the explanation of there being such a difference in the temperature at which water will boil? For instance, in a nearly perfect vacuum it will boil at 100 degrees temperature, while if exposed to the atmosphere it is said to boil at 212 degrees. A. The atmosphere exerts a pressure of 14.7 pounds per square inch to prevent that agitation of the water called boiling, while in the first instance you cite there is a pressure, reckoning from vacuum, of no more than one pound. The greater the pressure on the water the greater the heat required to agitate it and permit the vapor to escape. Thus under an absolute pressure of 1,000 pounds per square inch, water would not boil until a temperature of nearly 550 degrees was reached.

114. J. C. R. writes: I do amateur work and have trouble in getting the right temper on small tools. I heat by a special gas heater and draw by the color. What troubles me is I draw two tools to exactly the same color and one will be soft and the other hard. Why is this? A. You appear to assume that color is, primarily, an evidence of some corresponding degree of hardness; It is nothing of the kind, except by accident. It is simply evidence that the steel has been heated to a degree corresponding to the color. The color will show the same if the steel has not been hardened. If then you do not, in the first place, heat the steel hot enough to harden it as hard as you want it, the color test will be deceptive. If you first harden the tool a little harder than you want it, you can satisfactorily judge of the temper in drawing by the color. Perhaps in your effort to avoid heating, to harden, too hot, you do not heat hot enough.

115. J. C. B. writes: I am starting in to provide myself with a complete set of standard steel mandrels. I want them to be strictly true, but when I harden the ends the centers go out a little. I am afraid that after considerable use the centers will become round, but that the mandrel will not be quite true. What I want to ask is, if there is any way to keep the centers perfect in shape in hardening, and if there is anything you can suggest as to shape of ends of mandrel? A. There is no way that we know of in which the ends can be hardened with the certainty of leaving the centers true. We would suggest that you finish the ends about as shown in the sketch, roughing out



the body of the mandrel. Then ream the centers a little deeper if not deep enough, and harden the ends. Now catch a piece of copper in the chuck of a speed lathe if you have one, if not in a chuck of an engine lathe, and turn the end exactly to your center gauge. Keep one center on the tail center of the lathe and screw the other on to this copper lap, charged with oil and fine emery. In this way you can very easily bring the centers to exact truth, of course turning up your lap if it wears out of truth. Then finish the body of the mandrel; use a fair-sized center drill, say ⅛ inch for a 1-inch mandrel.

116. W. J. writes: I have a new Penberthy injector, and also a feed water heater, and I want to get the best results from both. Our water supply is from the city service, 90 pounds pressure to the square inch, and we carry 100 pounds steam in our boilers. I connected the city water direct to the feed water heater and had it pass through the heater and discharge into a barrel. The water level in the barrel is governed by a float valve. I found the injector would not lift the water at a temperature of 180 degrees, which it has when leaving the heater, so I connected the city water directly to the barrel and discharged through the heater from the injector. Is that the best way? A. There are two ways in which the inability of the injector to lift the hot water from the heater might be got around. One is by the plan you have adopted, and the other by piping direct from the water supply to the injector and feeding through the heater. In this case it would probably be advisable, perhaps quite necessary, to use a reducing valve in the supply pipe. Your plan is all right.

117. L. B. W. writes: (1) Is it necessary for one who desires to become competent to take charge of a large stationary or marine engine plant to serve an apprenticeship at the machinist's trade? (2)Are the engineers in the United States navy required to have served an apprenticeship at the machinist's trade? A. To both questions, no. On all sea-going vessels, whether naval or marine, the chief engineers are almost universally good mechanics, who are quite competent to superintend repairs of machinery and to do the work themselves, if occasion requires. In the instance of the transatlantic liners, the chief engineers are very generally men who have worked for some years at constructing and erecting marine engines. We should not advise any young man to enter the marine service with the expectancy of getting up to any very desirable position as engineer, unless he has learned, somehow, to handle the tools of the machinist; this he cannot do in any way so well as by serving a regular apprenticeship, and for his purpose preferably in a shop where marine engines are built. The first engineer in the navy should be an accomplished mechanic, well up in the theory of the steam engine and in applied mechanics. Good stationary engineers are not so commonly machinists, but they generally know about what a machinist can do.

118. The McG. E. Co. writes: We have a load of about 65 amperes up to 12 o'clock and after that the load is very light—not over 8 amperes. Would you advise us to put in a condenser, and would we save more coal by so doing and running our 115 horse-power single cylinder engine all night, instead of running the 115 horse-power until 12 and then changing to a 25 horse-power and small dynamo? A. Whether it would be more profitable to use a condenser in connection with the large engine than to use the small engine after 12 o'clock, would depend upon the number of hours you run at full load. The use of the condenser would save a considerable amount of coal during the heavy load run, but, on the other hand, it is probable that the small engine and dynamo would require much less coal for the light service than the large engine would. If the run at full load is long enough the saving in coal would be sufficient to more than offset the excess of consumption with the large engine during the hours of light load. The best arrangement, however, would be to install a storage battery. This could be charged during the hours of full load, and then after 12 the power could be shut down and the battery would take care of the light load. In this way the total coal consumption would be less, and in addition there would be a saving in wages, as the battery will run without attendance. This plan is being very extensively adopted by the larger light and power stations. and with very good results.

119. D. D. F. writes (from Nebraska): I have invented and am trying to build a small high speed over-hung crank steam engine. Our tools here are not such as are used in the East for building steam engines, and I can neither get journals of shaft or crank pin round, and when I get the crank pin driven into the crank it does not line correctly with the journals of the shaft. The result is that both journals and pin heat. I am aware that it is the practice to grind journals and pin round, but I have no machine for this purpose. Can I do any better than I am doing,

that is, turn them as round as possible with the old lathe? For the crank I bore it for the shaft, then move it on the face plate and bore for the pin; then I put the crank on the shaft and key it, and drive the pin (I have no press), but the pin never comes true with the main journals. The engine is all right in every other respect, except in the heating of journals and pin. The main journals are one-half the diameter of the cylinder and their length is twice their diameter. The pin is one-quarter the diameter of the cylinder and about ⅜-inch longer than the diameter. Are these proportions good and can you suggest any way by which, without too great expense, I can get the journals and pin for one engine round and in line? I propose to run with 90 pounds boiler pressure? A. Your proportions for journals and pin are as good as the best, and by the exercise of a little patience and by good mechanical horse sense you can get them into condition equally good. Turn the journals as well as you can with the old lathe. Have the centers accurately reamed to fit lathe centers. Make a casting that will take the place of the tool post and in which you can mount a shaft which will carry a small emery wheel made for such purposes, and which any manufacturer of or dealer in emery wheels will furnish you, and a small pulley to drive it. Drive this emery wheel, from the line shaft, or any other convenient shaft, at a speed recommended by the manufacturer. Harden the live center and grind both centers if you have the facilities for doing so. Probably you do not have the facilities, in which case ream three or four holes in a thick piece of lead with the same reamer the centers in shaft were reamed with. With the centers in the live spindle, and the centers in piece of lead for a lap, using fine emery, lap both lathe centers to get them round and in perfect shape. Screw down the head spindle box caps so as to make a dead center of the live center. Mount the shaft (with no dog) in the lathe and arrange to belt direct to it (no pulley is required) from some shaft, giving it a slow motion of revolution in a direction opposite that of the emery wheel. You now have a dead center grinding machine, and a very good one. Do not crowd the emery wheel against the journal. Let the touch be light and the feed coarse. In this way grind both journals precisely as if you were turning them, not changing ends with the shaft during the operation of grinding both. When you are through, if you have exercised care and good judgment, you will have two journals that are both round and in line, one with the other. For the crank bore the hole for the shaft the right size and the hole for the pin 1-16-inch small. Force the crank to place and key it. Then lay the shaft and crank on something level—like a planer table—and mount a boring bar as indicated in Fig. 1, carefully lining it by the main journals of the shaft. In this way the hole for the pin

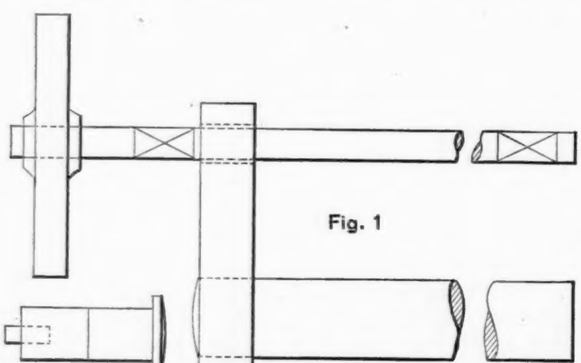


Fig. 1

Fig. 2

may be bored true with the journals. It will almost invariably be found to have been "thrown out" in keying the crank. For the pin finish both journal and part that fits the crank without changing ends in the lathe. If the stock is not long enough to turn a bit for dog, after roughing out a hole may be drilled—and tapped if thought advisable—and a plug for the dog inserted, as indicated in Fig. 2. This pin may be made long enough for attaching a small pulley, and both journal and part that fits the crank may be ground as the main journals are ground. Generally lead lapping the journal of the pin will round it up and make it right. The lead lapping consists in polishing it, with fine emery, between two half circles of lead that fit the pin. Since you have no press, when ready to drive the pin into the crank it is advisable to procure two pieces of iron a trifle smaller than the hole and alternately heat them at the forge and put them in the hole to expand the metal and make the hole larger. The pin will then require but little driving and will be more likely to go in true. By taking sufficient pains, even if your facilities are not the best, you can get your pin and main journals round and in line, thereby giving your engine a fair trial.

120. S. W. T. writes: I had always supposed that the pressure at the bottom of a boiler was greater than that at the top, owing to the weight of the water. Lately I have read of some experiments made on a marine boiler that went to show that there was steam instead of water at and near the bottom of the boiler, and the assertion was made that this indicated that there was no more pressure at the bottom than at the top; that is, than the gauge pressure above the water. This, it appears to me, must be true when there is steam between the water and the shell, but how is it when the water lies solidly against the metal? A. We presume the experiments to which you refer were some that seemed to indicate that in the water legs of some boilers, these legs being exposed to intense heat, the steam actually lifted and sustained the water. But the gauge pressure of that steam which sustained the water was actually greater than that above the water. Assume that you have a vertical tube, a, Fig. 1; b is a piece just fitting in the tube. Top of this is a weight, w. The end of piece b rests against the bottom of tube. The piece, b, may represent the water in a boiler, and the weight, w, the steam pressure above the water. The pressure against the bottom of the tube will be the sum of the weight of w, and the piece, b. If now an elastic piece, c, having no material weight, be inserted between the end of b and bottom of a, the pressure against the bottom will not be changed. The elastic piece, c, represents the steam pressure supposed to be between the bottom of boiler and the water. It is evident that the pressure against the elastic piece in Fig. 2 will be the same as against the bottom of the tube in Fig. 1. Therefore, if the steam at the bottom holds up the column of water, its pressure will be as much greater than that at the top as is due to the weight of the water. Suppose the depth of water is 3 feet; a column of water 1 inch square (1 square inch) and 1 foot in height will weigh a little less than one-half pound. The pressure at the bottom of the shell will be, say, 1¼ pounds per square inch greater than at the top.

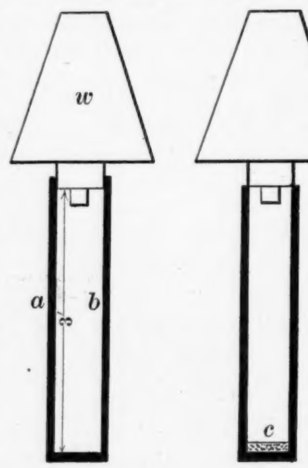


Fig. 1 Fig. 2

121. J. E. G. writes: I have two engines, the one 10" x 20" and the other 20" x 30", connected to separate shafts. The engines are neither of them heavily loaded. The boilers are old and we carry but 50 pounds boiler pressure. Both engines are non-condensing. What I propose to do is to add a condenser to the larger cylinder and operate the engines as a compound. They are only about 30 feet apart. I shall decrease the revolutions of the smaller engine and increase the revolutions of the larger one, so that the revolutions of the two will be the same. Can I work them that way, and what will be the saving? A. You would effect a greater saving by adding a condenser to the larger engine and doing all the work with that engine. The steam pressure is too low for compounding, and especially where one cylinder is of so much larger capacity than the other. To get good results from cylinders of such proportionate capacity the boiler pressure should not be less than 165 pounds, but this would not do in your case, as your engines are now underloaded. Running your engines disconnected you will never be able to succeed in making the engines make equal revolutions, making it advisable to use a large receiver. If you will compound the engines, bring them both together, and connect at right angles to one shaft, with a fly wheel pulley between the engines, and arrange your shafting to correspond with the requirements; you can get about the same work from each cylinder by putting an independent adjustable cut-off on each and adjusting

them by the aid of the indicator. Use a throttling governor on the high pressure cylinder and adjust the cut-off to be as short as possible, and leave a sufficient margin for the governor to control the speed. You can spend considerable money in all this and in all probability not get so good results as with the large engine alone, as suggested.

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FRESH FROM THE PRESS.

The Indicator Hand Book. By Chas. M. Pickworth, Editor of the Mechanical World, Manchester, Eng. D. Van Nostrand Company, 23 Murray Street, New York. Vol. I. Price, $1.50.

This is the first one of two volumes that the author is preparing upon the indicator. It treats upon the construction and application of the indicator, and the second volume will take up indicator diagrams. There are so many books upon the indicator that at first thought one would question whether there was room for another one. Mr. Pickworth, however, has made the volume which we have received very complete, although it contains only 126 12mo pages. The construction of the indicator, with its errors in actual operation, are dwelt upon at length, and there is a full treatment of various forms of indicator reducing gear. This chapter is the most complete that we have seen upon the subject. It not only explains the construction and proportions of various gears, but points out their sources of error. Much of the matter in this book will be familiar to American readers who have followed American discussions upon the subject in various technical journals. The author has drawn freely from this source, and will doubtless thereby make his work of considerably greater value to English readers than other English works which have yet appeared.

ADVERTISING LITERATURE.

THE STANDARD SIZES FOR CATALOGS ARE 9 X 12, 6X9 AND 3½ X 6 INCHES. THE 6 X 9 IS RECOMMENDED, AS THIS SIZE IS MOST LIKELY TO BE PRESERVED.

The International Correspondence Schools, Scranton, Pa., have issued a small but comprehensive pamphlet telling what correspondence instruction is, and giving a history of this system of instruction as developed by this company at Scranton. Samples from some of the instruction and question papers are included.

The Harrison Safety Boiler Works, Harrison, Pa. Catalogue of the Cochrane Separators, 48 pages, 5¾x7½ inches.

The subject of separating oil, water, etc., from steam or other gaseous currents is treated in this pamphlet, and the various forms of apparatus made by this company for the purpose are illustrated and described. Tests are quoted and lists of all the sizes are included.

The Massachusetts Fan Co., 53 State Street, Boston, Mass. Catalogue B of the Davidson Fan Wheel, 27 pages, 5x8 inches.

Various sizes and styles of fans for ventilating purposes are illustrated, ranging from small fans driven directly by electric motor, to those of large size for heating and ventilating buildings. A number of important buildings in which these fans are used are illustrated.

Loring, Coes & Company, Worcester, Mass., send up a small but interesting advertising circular entitled, "What the Dons See." It represents, presumably, a port hole in Morro Castle with the armored cruiser "Brooklyn" seen in the distance. A full view of the cruiser appears when the front part of the circular is folded back. They advertise to make knives to cut everything except Spaniards.

The Joseph Dixon Crucible Company, Jersey City, N. J., issue a pamphlet upon Graphite as a Lubricant. This is an enlarged edition of a similar pamphlet published last year and is well worth the careful reading of any engineer, giving, as it does, much practical information about the use of graphite. It is stated that copies of the pamphlet, as well as samples of graphite, will be sent to any one applying for them.

Patterson, Gottfried & Hunter, Ltd., 146-150 Centre Street, New York. Catalogue of machinery, metals, hardware and tools, 32 pages, illustrated, standard size.

The tools listed in this catalogue are those made by the Springfield Machine Tool Company, The Whitney Manufacturing Company, and the Builders' Iron Foundry, and include a variety of machine tools and grinders, as well as smaller supplies.

The Ingersoll-Sergeant Drill Company. 26 Cortlandt Street. New York. Catalogue No. 32 of Air Compressors, 6x9¾ inches, 96 pages, illustrated.

This catalogue illustrates and describes very fully the air compressor product of this factory. It is an interesting and complete catalogue of this line of machinery and will not only be desired by the purchaser, but will prove of value to any who are interested in this class of machinery. It has many well-executed illustrations, not only of individual machines, but of plants where they are in use.

J. E. Snyder, Worcester, Mass., catalogue of improved upright drilling machinery, 52 pages, 6x9, standard size, illustrated.

The 1898 edition of this catalogue shows a number of improvements and several new styles and sizes of drills. The sizes listed range from 20 to 36-inch, and the various styles include wheel and lever feed drills, and power feed with and without back gears. Also, one style is shown with a revolving table driven by power and another with compound table having rectilinear feeds.

The Lidgerwood Manufacturing Company, 96 Liberty Street. New York, have issued advance sheets of their forthcoming catalogue of the Lidgerwood Safety Derrick Engine. These sheets compare, by nicely executed drawings, the relative ease of operating an ordinary double drum hoisting engine, where one drum hoists the load and the other operates the boom, with the Lidgerwood Safety Engine, which not only makes provision for safety, but for ease of operation.

We have received catalogues of the State Agricultural College at Logan, Utah, and the Bradley Polytechnic Institute, Peoria, Ill.; and from the West Virginia University an announcement of the courses for the summer quarter. Doubtless any of these institutions will send their catalogue upon application.

MANUFACTURERS' NOTES.

An error in the July advertisement of the Scranton Correspondence Schools made it appear that "more than 25,000 students endorse these schools and testify to the benefits received from their quarter;" whereas the number should have been 45,000. We have often had occasion to commend these schools to our readers, as offering the most thorough means of acquiring a technical education at home that exists to-day.

A combination in the rheostat business has just been conciliated by the consolidation of the Cutler Hammer Manufacturing Company, of Chicago, and the American Rheostat Company of Milwaukee. The business will be carried on under the name of the Cutler Hammer Manufacturing Company, at 70 to 80 West Jackson Street, Chicago, Ill.

The Waterbury Farrel Foundry & Machine Company, of Waterbury, Conn., are full of business, and say that all the Waterbury manufacturers are similarly situated, more mechanics being employed in that city at present than ever before in its history.

The Cling Surface Mfg. Co., Buffalo, N. Y., manufacturers of the Cling surface belt dressing, write that they have lately received an order from the DeLamar, Nevada, Gold Mining Co., for three barrels of Cling Surface as the result of an order and the use of a sample can.

Dreses, Mueller & Company, Cincinnati, O., have added 1,500 square feet of basement to their capacity for smith shop, store and wash rooms. They have also added another story of 6,000 square feet, making a total floor space of nearly 15,000 square feet. They now have two distinct departments, one for building their new radial drills and the other for their screw and turret machine business. The changes at their works included also a new brick front and a removal of the offices and drawing room to the second floor. They report business as very good, but foreign orders constituting about 80 per cent. of the whole. Mr. Dreses intends soon to make an extensive trip through Europe in the interests of their foreign business.